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SUBMARINE ESCAPE.

BY LIEUTENANT C. B. MOMSEN, U. S. N., MEMBER.

In reviewing submarine accidents in the past, we find record of only three cases of successful rescue of crews from submarines by outside forces alone. The first is the case of the *U-3*, a German submarine, which sank at the dock at Kiel, Germany. The *U-3* was sunk due to failure of the crew to close the after hatch, and twenty-five men were entrapped in the torpedo room. Cranes were used to hoist the bow of the *U-3*, and at least 25 men were rescued. The German rescue ship, *Vulcan*, was present at the time, but could not be used to assist in the rescue work. The second case was that of the Danish submarine *Dykerren*. This vessel was also raised by means of cranes, and five men were rescued. The third case was that of the *O-5*, an American submarine, which sank at Coco Solo, Canal Zone. The floating cranes *Ajax* and *Hercules* were used to raise the forward end and two men were rescued after having been imprisoned about thirty-six hours. There are many other cases where submarines have been

sunk and attempts have been made to raise them, but in each case, as in the case of the *S-51* and *S-4*, the attempts have been without results. This has been brought up to point out the futility of trying to raise submarines by outside assistance alone, especially when they have gone down in the open sea.

The idea of getting out of submarines by means of individual escape apparatus has been occupying the minds of submarine people all over the world for more than twenty years. Several types have been perfected, but all of them have failed in practical application. The Germans developed the "Draeger"; the British, the "Siebo-Gorman"; and the Americans the "Diving Hood." The main difficulty with all of this apparatus was (*a*) excessive weight and size, (*b*) it was also complicated, and (*c*) no serious attempt was ever made to train submarine crews in its use. It was admittedly a gesture for the public.

After the *S-4* was lost with all hands this idea again came to the front and a great many people thought about it again. I happened to be the Line Submarine Officer on duty in the Bureau of Construction and Repair in the Navy Department and I felt that the matter of escape from submarines was a possibility and that an apparatus could be devised which would be free from the objections of previous apparatus designed for the purpose. The Chief Constructor, Rear Admiral J. D. Beuret, (CC) U. S. N., assigned me to this job of following this line of thought and investigation. I went to the Washington Yard and there became associated with the organization which developed the present "Lung." Chief Gunner C. L. Tibbals was in charge of the Deep Sea Diving Experimental Unit and was equipped with knowledge of all diving problems, apparatus for conducting tests, and men who knew their business in the diving game. Mr. Frank M. Hobson of the Bureau of Construction and Repair, an engineer, supplied knowledge of all known diving apparatus, a thorough knowledge of the "ins and outs" of equipment and material and how and where to get it. He also contributed to no small extent in the intricate problems of mechanical details of construction. Lieutenant Commander G. H. Mankin, M. C., U. S. N., volunteered his services as medical adviser. His experience with divers as well as with submarines and submarine people, together with his level-headed-

ness and common sense contributed to a large degree to the success of the entire project.

With this talent, and added to it the nerve and ability of such divers as Chief Gunner Laughman (of *F-4* fame where he descended to 306 feet), Chief Torpedomen Eiben and Kalinoski, the problem was thoroughly analyzed. Various types were developed and tried and failed. They did not meet the requirements of being small, light and simple of operation. We finally stripped all loose gear, such as oxygen bottles, air flasks, reducing valves, etc., and came to the type from which the Lung type was made.

In the early stages of development of this particular apparatus, the Navy standard rescue breathing apparatus was used. This was cumbersome, and we found it difficult to breathe under water. A study of the reasons for this difficulty disclosed that the air was being exhaled through the soda lime canister. We reversed the train of flow of air, breathing directly into the bag and inhaling through the canister. This improved breathing and the reason is that exhaling into the bag is necessary to force the air down to about 1 foot, or against a pressure of about ½ pound. It also requires about ½ pound of pressure to force air through the soda lime canister. In the first instance, we were breathing against a total resistance of about 1 pound, which was beyond the capacity of the breathing muscles. After reversing the train of air, breathing was found to be very easy. Departing from the rescue breathing apparatus, which has both air flask and oxygen flask, and other complicated attachments, we constructed a large bag, holding about 1000 cubic inches of air. This, of course, added considerable buoyancy and we found it required about 25 pounds of lead ballast to hold the subjects down. It was necessary to reduce the size of the bag, and we were influenced in our choice of the size by the natural example of the human lungs, which have an average capacity of about 250 cubic inches. The first lung that was made was the one that we made all of our tests with up to 100 feet.

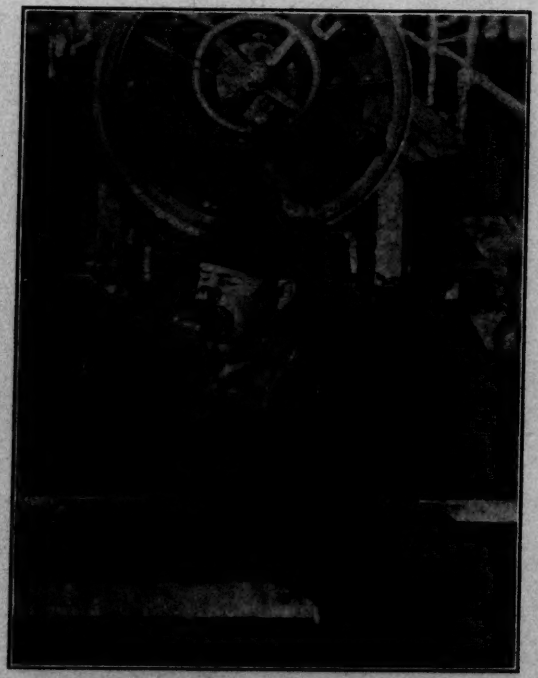
This apparatus has a rubber bag with about 250 cubic inches capacity. The rubber is specially treated and constructed so that it will not snag or tear, and is protected against oil, oxygen and salt water. The very best of material is considered essential in the construction of apparatus of this nature because of its intermittent

use and the undesirability of having frequent replacements. Inside this bag is placed a small canister with a capacity of from ¼ to ½ pound of soda lime. The soda lime is used for air purification, removing the CO_2. Two tubes leading from the top of the bag to the mouthpiece are fastened together and enter the mouthpiece. In the chamber on the outside of the mouthpiece are two mica disc valves which guide the air through the left-hand tube on exhalation and the right-hand tube on inhalation. There is a valve between this group and the rubber mouthpiece itself, which can close off the air, retaining the air in the bag for use as a life preserver. The rubber mouthpiece is so constructed that it fits securely in the mouth and very little effort is required to make a water seal. The vertical strip fits between the lips and teeth and the two little lugs are gripped between the teeth. The top strap passes over the head, the middle strap around the waist, and the two lower clips attach to the clothing about the legs. The nose clip is tied on to the mouthpiece for the purpose of closing the nose and assisting the wearer to breathe through his mouth. With a little experience the use of the nose clip is unnecessary. There is no protection for the ears and eyes. At the bottom of the bag there is a flutter valve, which permits excessive air to escape and, at the same time, prevents water from flowing in. This valve serves to allow excessive air to escape which is built up by expansion of air in the body as well as in the bag.

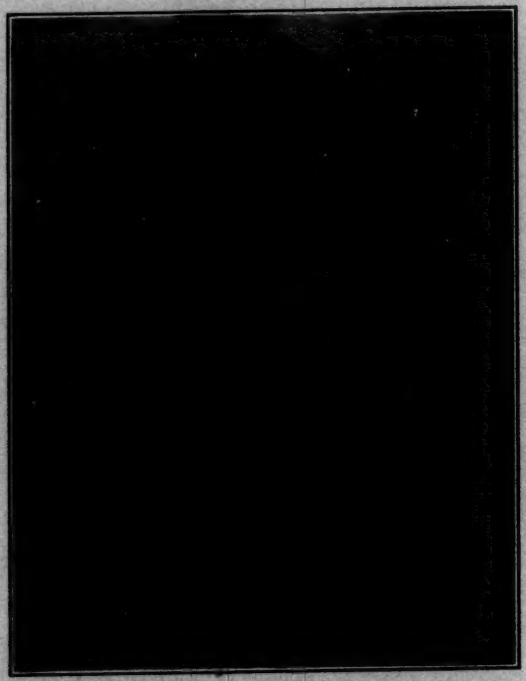
The first tests were conducted tests in the Model Basin at the Navy Yard, Washington, D. C., in 15 feet of water. All members of the experimental crew ran this test for about 80 days. The men were observed very closely and breaths were counted. Each man would put on the apparatus, walk down a ladder into the water and then move about on the bottom, rise or sink on a line or just sit on the bottom, but breathing, of course, all the while. These men remained under water for from two to six minutes. The doctor examined them after the tests and found them in good condition.

The next step was to construct a diving bell. This bell consisted of a tub 2½ feet in diameter and 18 inches high, open end down. Two uprights leading down 4½ feet, secured to the sides of the tub 180 degrees apart, were connected by a cross piece of

Escape Lock Showing Ascending Line and Buoy.



Emerging from Motor Room Hatch.



EMERGING FROM ESCAPE LOCK.

ESCAPE LOCK.

wood 6 inches wide. The whole was then ballasted by attaching lead on the bottom of the uprights. The bell was lowered into the water in the mine tank, beginning at 20 feet. The men who were conducting the tests would stand on the cross pieces with their head inside the tub, and thus were able to breathe in this air space. Upon reaching the required depth, they put on the lung and came out, coming up the ascending line very slowly. Depths in this mine tank were increased to a final depth of 60 feet, which was the limiting depth of this tank. When we were all graduated from 60 feet, it became necessary to find greater depths of water. In the meantime the lung was tested in a pressure tank at the Navy Yard, Washington. This tank, about 10 feet in diameter and 12 feet high, was filled half full of water; pressure was applied by admitting compressed air in the upper air space, building up the pressure to correspond to the number of feet in depth required in the test. The subject would then put on the lung and get under the water, and the pressure would be released at the same rate that it would be released naturally on ascending through actual water.

We then obtained the *Crilley*, an old diving boat, and put our expedition on board, went down the Potomac River to a point off Morgantown, where there is depth of 110 feet of water. Two of us, Eiben and myself, went to the bottom by means of the diving bell in succession and returned safely to the surface by using the lung. The time required to come up averaged five minutes. In each test, two men went down — one tender, who stayed in the bell at all times, and the other acted as a "guinea pig," using the lung for escape.

The next tests were made in Chesapeake Bay, off Solomon Island, in 155 feet of water. Here three (Eiben, Kalinoski and myself) were lowered in succession to 155 feet and made successful escapes to the surface; the time for coming up in this case averaged about seven minutes.

The *S-4*, which was sunk off Provincetown in collision with the Coast Guard ship *Paulding*, was recommissioned as a submarine experimental hulk by authority of the Secretary of the Navy. All machinery and apparatus was installed in the control room and battery compartment. She had pumps, compressed air, storage

battery, kingstons, vents, periscope and other apparatus for controlling the vessel on stationary dives. There was no propulsive power. The *S-4* was towed to Key West. On board were a voluntary crew of picked men and a Commanding Officer (Lieutenant N. S. Ives, U. S. N.). Attached to the vessel were Lieutenant Shugg, Construction Corps, and Chief Gunner Tibbals, a deep sea diving expert and a crew of fourteen men. At Key West the *S-4* was submerged in water at depths varying from 40 to 202 feet. Escape was made from the motor room, torpedo room, and engine room at various depths, up to 100 feet.

In flooding the compartments, the following procedure was followed. The hatch strongback was released first or the hatch was released, the water was then admitted into the compartment through a nine-inch flood valve, which had been previously installed and was controlled from the Control Room. The men stood around in the compartment waiting for the water to come in, and, of course, with the entrance of the water the pressure began to build up. As the pressure inside began to approach the sea pressure outside, the hatch began to leak—at first a few drops and then these developed into a stream. The stream increased until it was about eighteen inches in diameter. It was easy to observe that the hatch was open about six or seven inches on the free side and that the pressure on the inside just balanced the sea pressure on the outside and held the hatch in this position. As the water rose in the compartment, the excessive air escaped out of the hatch until the height of the water was just level with the bottom of the hatch skirt. This hatch skirt extended down into the compartment about three feet. The water continued to rise inside the hatch skirt until everything was quiet. It was then necessary to crawl up the ladder and throw the hatch wide open and return to the compartment. The next step was to pass up a small cork buoy attached to a line, the size of signal halyards. When the buoy watched it was easily determined by the change in the pull on the line. The line was then made fast and the men were ready to escape. On this line, measuring from the buoy every ten feet, were attached wooden balls, down to 70 feet, so that men coming up could determine the distance that they were from the surface. The men then put on the lung, charged up with oxygen, ducked

down under the hatch skirt and allowed themselves to rise on this line to the first stop. At the stops they were required to remain for short periods for decompression. The wooden balls were not placed lower than 70 feet from the surface since that is the first stop for any depth exceeding 140 feet. For depths less than 140 feet the first stop is approximately half way to the surface. I did not notice any difference in escaping from depths greater than 60 feet. The 200-foot escape had no more sensations than 100 feet, and I do not believe that the men could tell, with reasonable limits, at what depth they were unless they could see a pressure gauge.

At a depth of 200 feet the water was very clear — a dark blue,— and at that depth the light in the water makes a picture resembling starlight. Nearer the surface, say at about 100 feet, the light gave the appearance of moonlight, and the visibility was from 25 to 50 feet. Of course, as the surface was approached, the light increased to sunlight.

While the fish in the vicinity of Key West are very large and vicious, we did not experience any difficulty from them during these tests. The precautions that were taken were, first, to run an oscillator at frequent intervals, and, second, to have a train of air bubbles escaping continuously from the submarine. We had been told by various people that these two things would frighten the fish. Whether or not they do, we are not in a position to state, but we were very gratified at not seeing any of them.

The three types of escape which we made from the *S-4* were as follows:

(*a*) The motor room: A circular hatch coaming leading down from the hatch trunk about three feet into the compartment and a regular horizontal hatch overhead.

(*b*) The torpedo room: A partial bulkhead extending down about two feet from overhead and nearly across the compartment, which entrapped an air pocket in the upper part of this compartment from this partial bulkhead forward, and a 45 degree plain hatch.

(*c*) The escape lock: This is a trunk about seven feet high and four feet in diameter, placed on top of the engine room hatch. Entrance to this trunk is through the engine room hatch. There is a side door escape on the after side of the trunk, the top of which

is about 18 inches from the top of the trunk. On escaping from the escape lock, it is necessary for two or three men to enter the lock, closing the engine room hatch and flooding the lock from sea. As the water enters, pressure is equalized, and when the water ceases to flow, the side door is open, the buoy sent up and the men step out on deck and thence on up to the surface.

While in all our escape work and test work with the lung, oxygen has been used for filling the bag, it is believed that the same thing can be accomplished using the air in the lungs for inflating the bag. It did not seem desirable to confuse these tests by trying out this method of inflation, but we intend to proceed with further tests to determine to what extent oxygen can be eliminated. We have already made one test to 110 feet, using five minutes for decompression in coming to the surface. It is generally known that the value of oxygen percentage increases in proportion to the atmospheres of pressure in which you are breathing,—for instance—10 per cent of oxygen in two atmospheres of air will give the equivalent of 20 per cent oxygen; and 5 per cent of oxygen uder four atmospheres of pressure will also give the equivalent of 20 per cent oxygen. Consequently, when the bag is inflated from the lungs it receives air containing about 17 per cent oxygen. Of course, as this air is rebreathed, the oxygen percentage is lowered with each breath, but since the whole is under pressure, the effect of the oxygen is that of more than normal. We believe that breathing under these conditions should cease at about 30 feet, and the subject should come directly to the surface from that point. There is, however, still a difference of opinion among the authorities as to the practicability of this procedure and a very careful series of tests will be conducted.

Training in the use of the apparatus is very simple, but is also extremely important. Without adequate training the apparatus will be of very little value to the submarine crew. The Department is contemplating building a tank for training purposes at the Submarine Base, New London, Connecticut. In the bottom of this tank will be constructed an exact duplicate of a submarine compartment, with standard hatches and fittings. There will be 100 feet of water in the tank above the compartment hatch. Men will be trained, beginning at shallow depths and culminating in an

actual escape from a submarine compartment through 100 feet of water. A man once trained in the use of this apparatus will probably never forget how to use it. We estimate that a period of 10 to 15 minutes each day for three days is sufficient for instruction.

A matter of great importance has come to our attention during the research and development work of this apparatus, viz: the effect of CO_2 under pressure. We have found by actual tests that 2½ per cent of CO_2 in four atmospheres of air will render men unconscious in a very few minutes, whereas this same amount of CO_2 under atmospheric conditions is absolutely harmless. Thus a submarine which has been operating 8 or 10 hours submerged may have accumulated as much as 2 or 3 per cent CO_2, and if she is rammed and sunk, or has compartments flooded by any other means, the pressure will be built up to correspond to nearly that of sea pressure and may introduce a grave element of danger. It is going to be necessary to take steps against the dangers of CO_2 under pressure.

The Bureau of Construction and Repair is taking steps to equip all submarines with the lung, two for each man plus ten per cent for spares. The two end compartments will be fitted as retreat and escape compartments and each end will contain sufficient apparatus for the entire crew. This is done so that no matter where a submarine is struck or flooded the men may retreat to the bow or stern toward safety. Of course, if the submarine is damaged in two places simultaneously the men between those points will be in a bad way. Such an accident, especially in peace times, is hardly conceivable. The steps that are being taken seem to put the submarine man on a par with the surface ship sailor, that is, to put him on the surface to be picked up by another ship.

A PLEA FOR SOUNDER ENGINEERING COMPETITION.

By Roger Warde Paine, Lieutenant Commander, U. S. N., Member.

It has become well recognized through centuries of effort in the field of sport that if endeavor is to be incited and interest sustained — in other words, if it is to be a success — any form of competition must be governed by rules based on principles of such certain fairness that each competitor will be given a chance to win.

To meet this prerequisite two distinct forms of competition rules have been evolved, which may be classified as "scratch" and "handicap." The first class meets the demand when all of the contestants are equipped with the same tools, physical, mechanical, or both, and competition interest is sustained by the degree to which this demand is met. The second class of competition has been devised to meet the need when all contestants are not equipped with the same tools, and it becomes necessary to establish a common level by some system of weighting the high and boosting the low.

In this imperfect world of ours, handicap competition is much the commoner form, and a great deal of thought and time has been put into devising methods of weighting which will incite endeavor and sustain interest in any particular kind. Of such is the Naval Engineering Competition which falls naturally into the handicap class. There could be no "scratch" competition between an old battleship commencing to show signs of wear and tear, and a modern one equipped with the last word in fuel-saving devices. The new ship must be weighted down in her fuel consumption, or the old ship boosted in hers before any semblance of comparison can be found.

That the Engineering Competition has been a great success, no one can honestly doubt. In its twenty years of life it has returned

to the Treasury a vast amount of money which would otherwise have been spent extravagantly and wastefully in the extra expenditure of fuel to accomplish exactly the same mission. It has taught our engineering personnel to analyze expenditure, eliminate waste, and to anticipate trouble. It has reduced the cost of maintenance of the Navy in peace-time, and it has increased the cruising radii and reliability of ships in time of war.

But in the past few years its machinery has given evidence of the need for an extensive overhaul. There have been unpleasant noises in the gears with a missing cylinder now and then, and unless something positive is done in the way of repair it seems only too probable that the whole machine will be swallowed up in that horrible condition known as innocuous desuetude. The Engineering Competition is too valuable to the Navy and to the country to allow this to happen.

Considerable thought and study indicate that whatever trouble exists lies almost wholly in the handicapping system which is now employed. That it violates the fundamental principles enunciated as governing any form of competition must be apparent to anyone casually listening in on any one of the perennial ward-room arguments on the competition, or to anyone interested enough to sound out the consensus of opinion of the fleet engineers who are compelled to work under it.

In recent years there have been a number of striking cases in point. One ship becomes conspicuous because of her inability to keep position in formation, but her high "multiple" enables her to take penalty after penalty without altering her very high position in the competition although ships below her are obviously running quite as economically and operating with perfect reliability. Another stands very high in the competition and on an even basis with other ships of her class although during a period of three months while they were steadily cruising in tropical waters in the most grueling test to which the fleet has been subjected in recent years this ship lay at a dock in Navy Yard completely out of the competition. Another just fails to make first place in two successive years, eased out each time by different ships under circumstances which, whether rightly or wrongly, completely disheartened her personnel, the high pitch at which she had been

operating being indicated by the drop she took to the bottom of the list when she gave up during the third year.

Improperly handicapped, the performance becomes perfunctory at best, and at the worst, defeats its own ends. For, under the present scheme of things, a ship doing poorly in competition through one year, works to its own advantage in future competition if instead of doing poorly, she does as poorly as possible. Her allowances for any one year are based on her performances for the years immediately preceding, and if these performances have been poor in comparison with her true capabilities, her allowances become so artificial that a normal display of engineering talent, a little assistance from the deck, and some hard work, will produce a tremendous "multiple" which does not in any way reflect her comparative efficiency with ships which have plodded along with little variation in efficiency year after year.

This, if it be clearly understood, accounts for a good part of the criticism which may be heard with regard to the competition. It accounts for the often apparently deliberate switch of the lead from class to class. It accounts for the attitude of many engineer officers who test out their new allowances for a month or so, and seeing that the die is cast inevitably against them, continue on doing their best without enthusiasm, or flatly let things go, according to their dispositions. It means, to sum up, that the trouble with the Engineering Competition today lies very squarely with the "allowance" method of handicapping.

An "allowance" may be defined as the Department's estimate of the amount of fuel oil in gallons, or of coal in tons, which a vessel should average in consumption over the course of a year when operating at a certain speed with a given set-up of her engines. For the purposes of this discussion, at anchor allowances, and the special allowances for water, electricity, and unusual conditions will be ignored, as they aggregate but a small percentage of the total fuel consumed by a vessel of the fleet in the course of a year. Allowances then will be understood to mean the allowances underway.

These are worked up by the Department and issued to ships in the form of tables covering every normal speed that the vessel may run on each of her engine combinations to the nearest tenth of a

mile on her clean-bottom revolution per minute curve. The data for these tables is taken from curves which are prepared in the Office of Fleet Training, the curves being drawn through points plotted from data submitted by the ship herself and representing runs which have lasted long enough to insure a fair average fuel expenditure per hour being obtained. These points are collected from data submitted during the previous three years of performance, and, as might be expected, form a "milky way" across the coordinate sheet through which the curve must be "faired." Each speed is represented by many widely varying points showing fuel consumed because it is rare if ever that a ship would ever travel at the same speed with the injection temperature, the auxiliary power requirements, the bottom cleanliness, and the displacement exactly the same on two different runs. Yet all of these factors affect her fuel consumption considerably.

It will be observed then that a tremendous amount of work is represented in the production of the allowance curves and also a great deal of care and thought. Anyone who has "faired" a curve through many widely varying points will realize the good judgment which must be exercised in producing it and the considerable doubt which exists even when good judgment is used. To the curve so produced, the Department may apply a factor of reduction to incite improvement, and other factors which will tend to force ships to use the most economical engine combination when two are available at the same speed. Such factors, of course, introduce elements of doubt quite beyond those inherent in mechanical drawing.

The points through which these curves are drawn are themselves rather ephemeral things, and open to all manner of question and criticism. They represent "runs" at hourly speeds not varying more than two miles per hour. In any hour, however, the speed may have been changed from zero to a maximum with a consequent average speed for the hour which is quite low, yet accompanied by an unstable plant operating condition as costly in fuel expenditure as an hour at the highest speed. This is a frequent occurrence in maneuvering, and in gunnery exercises. It is a recognized error in the present system of handicapping which cannot be corrected without a tremendous amount of additional work

in the preparation of the Form H, and a consequent considerable increase in the chance of clerical error.

It will be seen also that a basic assumption is made in the assignment of allowances which is manifestly not well-founded, and responsible for what must amount to a large error in this method of rating. It is assumed that because a ship has used a certain quantity of fuel at a certain speed in one year that she should necessarily average the same amount in the next. It frequently happens, however, that in one year the points representing some particular speed have all been obtained in tropical waters while for the following year every point at that speed has been obtained in northern waters with a great difference in the over-all plant efficiency in consequence. One year may represent twice the mileage of the following year, the high mileage year meaning long sustained cruising where the plant has been settled down to its maximum economy, while the low mileage year means short runs, unsteady conditions, much getting underway and coming to anchor with consequent poor economy. As the needs of the Navy cannot and should not be subordinated to the Engineering Competition this variability of performance must be accepted, but it certainly seems unfair to penalize a ship because of it as very frequently happens.

History has demonstrated that as a class, naval officers are ultra-conservative. They have received a military training which has inculcated respect for existing authority, for the written word, and bred a conspicuous deference to custom. The trait is a laudable one and a necessity in a military organization but it can be carried too far. It should never be allowed to retain on the statute books a set of rules which are known to be wrong. It probably accounts, however, for the fact that once having accepted the "allowance" method of conducting the Engineering Competition and recognizing its shortcomings, that we have continually tried to mend the "allowances," instead of junking that particular method of handicapping, in favor of one better suited to the needs of the competition.

In seeking some method of handicapping which could be substituted for the present "allowance" method, it has been found difficult to do more than outline the broad features of a logical

replacement method. The Department, with its voluminous files of data and performance, must be the only capable formulator of the rule book, and for this reason only the general principles of a replacement method are here suggested.

To start with — a naval engineering competition can have but one purpose. It must be designed so that it will promote a spirit of rivalry between ships which will have for its aim the reduction of the expenditure of fuel used by any ship to the minimum which its employment justifies, securing this economy in fuel consumption without loss in engineering efficiency. In such a competition two general conditions of operation must be recognized which together will cover the whole employment of any ship during the course of a year. The first condition is that "underway," during which the ship travels her yearly mileage; while the second condition is that "not underway," largely made up of the "at anchor" periods, and all fuel burned by the ship in the course of a year may be readily prorated to these two conditions accurately, and with but a moderate amount of paper-work.

If, now, an accurate account is kept of the time underway, and the mileage is computed from the revolutions made good, the mileage figure divided into the fuel burned underway, the result will be "gallons (tons) of fuel per mile in average for the year." This expression, in the last analysis, is the only figure of interest in the ship's underway performance. For instructional and comparative purposes within the ship it may be worth while to know and record the engine combinations used and the actual fuel expenditures at various speeds, but the sole purpose of the competition after all is to make each ship do its yearly average mile with the least possible expenditure of fuel, and everything else should be subordinated, or directed, to that end. In a like manner, the expression "gallons (tons) per hour not underway," readily and accurately obtained, is the final figure sought in the second condition of operation.

Taken together, the two expressions "gallons (tons) of fuel per mile" covering the ship's underway expenditure, and "gallons (tons) of fuel per hour" covering the ship's not underway expenditure, show her performance exactly in the form in which Congress, the Department, and the ship herself are most interested. They are expressions which are computed now for every ship and

which are published in the very forefront of the Engineering Performance Report — but they have no bearing whatever on the relative standings obtained in the Engineering Competition. It is the premise of this article that they are the logical basic data upon which the system of handicapping for competition purposes should be devised.

It will be granted readily that if a ship steams at the same average speed in two successive years, but that in the second year her "gallons fuel per mile" is five less than in the first, there has been an enormous aggregate saving in fuel oil to the Government, and the ship has made a definite and measurable increase in her efficiency through the second year. Similarly, it will be granted that if a ship averages an expenditure "not underway" during her second year which is five gallons per hour less than in the first year, she has again made a definite and measurable gain in efficiency. Both of these expressions can then be weighted in respect to the fuel burned in each condition, and combined to produce an overall factor of relative efficiency.

Since all of our ships are built differently with widely varying types of engines and auxiliaries they are essentially incapable of attaining the same consumption figures for equivalent duty and any competition we devise must pit each ship not against any other one, but solely against herself and her best performance.

If we pit any ship against her own best performance in "gallons of fuel per mile underway," and "gallons of fuel per hour not underway," each properly weighted, and then modify the percentage so obtained by one which may be called "percentage of relative freedom from penalty," we at once have a measure of that ship's engineering performance with which she can be given a standing in competition with other vessels of her class or type — and, very important indeed, engaged in the same kind of duty and locality — each vessel entered in exactly the same manner.

In order to properly prepare a formula which will embody these requisites, let us now consider the requirements of a 100 per cent performance. They are simply expressed, and two in number.

First: A ship must at least equal her best performance both underway and not underway.

Second: The ship must do this without penalty (as under present rules).

In case two or more ships should have 100 per cent, or perfect performance, which is readily possible under this method, the Greatest Improvement Factor Formula (later to be described) would be used to determine their relative order of merit.

ENGINEERING PERFORMANCE FORMULA.

Let A = total penalties incurred during the current year.

B = average gallons of fuel per mile corrected to the mean average speed, which is the ship's best performance in any year (including the current one).

C = average gallons of fuel per mile corrected to the mean average speed for the current year.

D = average gallons of fuel per hour not underway, which is the ship's best performance in any year (including the current one).

E = average gallons of fuel per hour not underway for the current year.

F = gallons of fuel used underway during the current year.

G = gallons of fuel used not underway during the current year.

H = F + G = total gallons of fuel used during the current year.

$$\text{Engineering Performance} = \frac{100 - A}{100} \left[\frac{\frac{B}{C} \times F + \frac{D}{E} \times G}{H} \right]$$

This formula is quite apparent and easy to figure. All of its factors may be obtained from data now taken by all ships. It says, in effect, that if the ship operates during the current year with its fuel expenditure per mile equal to or better than that of any previous year, then $\frac{B}{C}$ becomes $\frac{1}{1}$ and the fuel used underway gets its full weight. Similarly, if the ship operates not underway so that she equals or betters any previous yearly performance, then $\frac{D}{E}$ becomes $\frac{1}{1}$ and the fuel used not underway gets its full weight. Thus, the numerator of the bracket fraction becomes F + G, or H, and the bracket becomes 1.00. If there have been no penalties,

the penalty factor becomes 1.00 also, and the ship's performance is 100 per cent, or perfect. If she fails to reach perfection in any one of the three main factors, her rating will be less than 100 per cent by a percentage properly weighted with respect to the fuel burned in the deficient condition. Once having produced the rating which represents her performance *in competition with herself*, she is now graded with other ships with this percentage which becomes a relative measure of the amount of effort she has expended, as against that of each of the others, in the effort to reach the lowest fuel consumption attainable.

One feature of this method of rating, and one which cannot be too strongly accented, is the fact that the most a ship can get in multiple is 100 per cent, and she *can* get this for perfect performance against her own standards, year after year. There is no such thing as a multiple of 110 per cent, a figure which can be obtained under the present method and one which is quite meaningless, for if the allowances are not properly assigned the figure can be obtained without effort.

In order to complete the picture of the ship's performance in any one year, still another relative rating is required — as it is required under the present method. This rating is one which will recognize an exceptional effort on the part of some ship's personnel either in setting a new high standard of performance which shows a considerable saving to the Government in one year of operation, or which restores the ship to an economical and efficient status when, for one reason or another, she has been allowed to drop.

This rating is called the Greatest Improvement Factor, and the formula for expressing it takes exactly the same form as that for the Engineering Performance. It will be noted, however, that in this formula the comparison is made between performances of the current and the *previous* year, instead of between the current and the *best*.

GREATEST IMPROVEMENT FACTOR FORMULA.

Let A = total penalties incurred during the current year.

J = total penalties incurred during the previous year.

C = average gallons of fuel per mile corrected to the mean average speed for the current year.

K = average gallons of fuel per mile corrected to the mean average speed, for the previous year.

E = average gallons of fuel per hour not underway for the current year.

L = average gallons of fuel per hour not underway for the previous year.

F = gallons of fuel used underway during the current year.

G = gallons of fuel used not underway during the current year.

H = F + G = total gallons of fuel used during the current year.

$$\text{Greatest Improvement Factor} = \frac{100 - A}{100 - J}\left[\frac{\frac{K}{C} \times F + \frac{L}{E} \times G}{H}\right]$$

Unlike the Engineering Performance expression, this formula gives an answer which may be greater than, equal to, or less than 1.00. Following the reasoning employed to show the application of the other formula, it will be seen that a ship obtaining 1.00 has repeated her performance of the previous year without gain or loss in efficiency. If the factor is greater than 1.0, she has improved, the decimal accurately representing the percentage of improvement; while if the figure is less than 1.0, the decimal represents a percentage of decreased relative efficiency.

A little study of the two formulae will show that the ratios which they express are simple, self-explanatory, and that the figures used are all collected now and are available in Reports of Engineering Performance for years back if comparison is desired. This data has been taken for purposes other than Competition computation, but it is an interesting commentary on the whole question that its summarization occupies the first pages of each yearly Report of Engineering Performance.

The only factors requiring explanation and presenting any novelty are B, C, and K in the two expressions. As they are not believed to have been advocated for comparative purposes previously, and as they are derived from the only data employed which can be manipulated to any degree whatsoever, they will now be discussed at some length.

They represent "average gallons of fuel per mile corrected to the mean average speed" for three different periods; namely, for the current year, for the previous year, and for the year of best performance. It should be noted that the last period may or may not be identical with either one of the first two. Also, since the factor for the previous year, and for the year of best performance become matters of record under this method, that only one new derivation is required each year — the factor for the current year. It is produced in the following manner.

It has been noted in studying the past performances of various battleships that their average speed from year to year — obtained by dividing the total mileage for the year by the number of hours underway — has not varied in the last six years by more than two miles over, or two miles under, a speed of eleven miles per hour. Because it affords a definite reference point in the case of battleships, this speed of eleven miles per hour has been selected for the purposes of this article as their "mean average speed." Other classes of vessels will doubtless yield an equally positive reference speed, and will certainly do so if classification for competition recognizes not only similarity of type but similarity of general employment and locality as well.

As part of the Confidential Supplement to the Rules, the Department would issue to ships a set of curves — one for each ship — similar to that in Figure 1 which is typical of that for any battleship. The ordinates of each curve would be "gallons of fuel per mile," and the abscissae "miles per hour," so that it is actually a limited portion of the fuel consumption curve of a vessel on her most economical engine combination. It should be noted that the data for producing this curve section is already available in F. T. P. No. 59, Cruising Radii of U. S. Naval Vessels, it can be produced very accurately by taking long-sustained Form H steady cruising data, and every engineer officer alive to the necessity keeps such a curve up to date now for purposes of comparison. It will further be observed that once having drawn the curve, the ordinates are erased preserving only the scale, as it is the slope of the curve only that is of interest in the computation.

With this curve at hand, the ship proceeds as follows (use the typical data given in the example, and refer to Figure 2):

U. S. S. BATTLESHIP.

Miles steamed in the current year	20,000
Hours underway	2,000
Average speed for the current year	10
Prick this point in on the curve as A in Figure 2.	
Fuel used underway in the current year (gals.)	1,400,000
Miles steamed in the current year	20,000
Average gallons per mile	70

Mark the ordinate passing through the point A in Figure 2 as 70, and number the other ordinates to a scale in gallons as indicated.

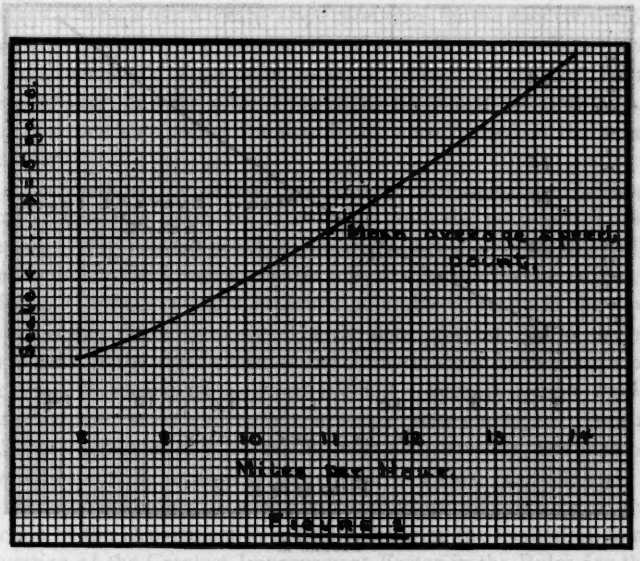


FIGURE 1.

The answer required is the value so assigned to the ordinate which passes through the "mean average speed" point (11.0), which happens to be 73 in this example.

The average gallons of fuel per mile corrected to the mean average speed, for the current year = 73. This becomes C in the two formulae, B and K have been similarly produced in previous years and are a matter of record.

Two assumptions are made in producing this factor, both of which are logical and borne out by past performance.

First. It is assumed that the *slope* of the fuel consumption curve of any vessel, at or near the point of mean average speed, can be produced by the Office of Fleet Training with a fair degree of accuracy.

Second. It is assumed that the average fuel consumption per mile of any vessel follows the slope of the fuel consumption curve of her most economical engine combination fairly closely.

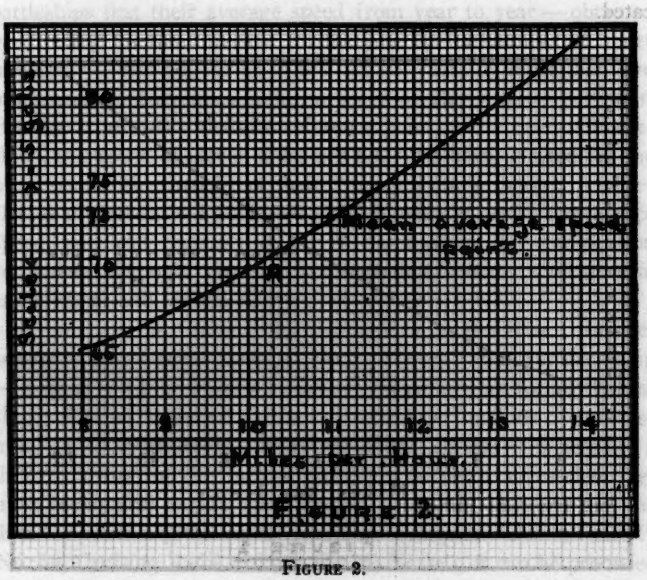


FIGURE 2.

The first assumption seems beyond question. Any engineer officer can draw in his current fuel consumption curve around the economical speed point (which is naturally close to the mean average speed point) with reasonable accuracy, and the Office of Fleet Training with just as much data available, and with trained draughtsmen, should be able to do it better. Actual test computations have shown that the error of using the slope of the tangent

at the point of mean average speed instead of the varying slope of the curve itself over the four-mile range, has but a negligible effect.

The second assumption is logical and sufficiently accurate for the following reason. If the ship during the entire period of a year did all of her cruising on the most economical engine combination, and at speeds between nine and thirteen miles per hour, the assumption would be strictly correct because the curves of unit and general performance are identical. A study of recent Form H data will show that a ship normally actually does expend all but about 5 per cent of the fuel expended in the course of a year between those speeds, so that the effect of operation at high or low speeds — which mean erratic slope conditions on the fuel consumption curve — must be negligible in amount. Whatever effect there is in producing a slope error must be reasonably constant from year to year, and the effect of a slight tilt would certainly be as fair for one ship as for another if all were figured in the same manner.

The details of the method which would be employed in putting this scheme into effect are quite aside from the purposes of this article. Any method selected, however, should provide the daily and monthly check and comparison now available to the engineer officer. Perhaps it might be found advisable to retain in the new scheme the collection of most of the data now taken. With the emphasis in the right place — the ultimate actual expenditure of fuel oil to accomplish a given mission — such collection would preserve the methods of daily comparison now employed. As the ultimate expenditure is so readily obtained, however, the method would essentially mean less paper-work — a most desirable end.

Before concluding, it seems appropriate to call attention to the action of the Department in radically changing the method of computation of the Greatest Improvement Factor in the Rules for the year, 1927-1928. Under the method which had been employed for a number of years, the formula used allowed changes in the allowances from year to year to weight the result which served to give a totally erroneous comparison. The method substituted compared the efficiency in burning fuel during the current and the previous year by requiring the computation of the "multiple" to be made against the allowances for the current year, and then quite inde-

pendently against the allowances of the previous year. A rather appalling increase in paper-work was thus placed upon the engineering personnel of ships, but a result was obtained which was strictly accurate, because allowances are ironed out of the comparison and it became a true comparison of fuel expenditure had conditions during the two years been identical.

The Greatest Improvement Factor Formula proposed in this article does the same thing in a much more direct way because it deals with fuel expenditures only. The labor-saving feature of the computation is enormous.

In conclusion, it should be observed that the method outlined in this article cannot be said to be a panacea for all evils. Its comparisons from year to year will be sensitive to changes of employment, locality, and unusual conditions, if they are of large degree. No competition can be devised which is not. But it will definitely bring out the effect of such changes so clearly that it is not inconceivable that after a few years of operation under it, that it will be possible to accurately factor such disturbing elements as injection temperature, average draft, and increasing age, so that their effect can be eliminated entirely. Such results are quite beyond the present system.

In the meantime, the method will give a strictly accurate relative rating in any one year, and remove from the minds of the engineering personnel of the fleet the feeling of doubt and dissatisfaction which now exists. If it can do this, it should be given a trial just as soon as the necessary machinery can be developed.

THE TODD PULVERIZED COAL BURNER.

TEST OF, ON A SCOTCH MARINE BOILER.

BY R. C. BRIERLY,* ASSOCIATE MEMBER.

EDITOR'S NOTE: In a monograph entitled "Marine Development of Pulverized Coal," which appeared in the August, 1928, issue of this JOURNAL, Commander J. S. Evans, U. S. N., and R. C. Brierly described the work on marine installations for pulverized coal burning which had been performed by the United States Shipping Board and the Fuel Oil Testing Plant. The present article deals with further work conducted from that time until the present.

FOREWORD.

During the summer of 1928, the Fuel Oil Testing Plant continued its test work with the Fuller-Lehigh Pulverized Coal Installation. Further developments were completed on two distributors for producing steady delivery of equal amounts of coal, with equal amounts of primary air, at equal pressures to the individual burners and boilers of systems comprising three Scotch boilers with a total of nine furnaces. The first of these distributors was a rifled cone which had the advantage of requiring no auxiliary power for operation; the second was a pneumatic device which necessitated the use of compressed air. The former consisted essentially of a truncated cone, flanged at both ends in order that it could be bolted in place in the primary piping before the three branch connections to the boilers, and again before the burner branch connections to the individual burners. Fins or plates in the form of helices were welded to the interior walls of the cone. The stream of coal passing through this distributor was thus caused to whirl—breaking up stream lines and stratification—and assuring more equal division into three streams than was previously effected.

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The latter type distributor consisted of a disc having a central circular opening the same inside diameter as that of the main line primary air piping in which it was installed; tangential orifices were drilled through the walls of this disc and were connected by lines, fitted with valves, to a supply of compressed air. The entrance of air under pressure through these orifices set up an air vortex in the coal stream and imparted a spiral flow to the coal particles; by varying the pressure in the orifices, the stream of air and coal could be spiralled in such a way as to cause practically equal flow to the branch connections. Figure 1 shows cross-sections of these distributors.

The rifled cone produced excellent distribution results at certain rates of coal firing, but was not entirely satisfactory throughout the desirable range of rates. Its ease of operation and ability to function without use of auxiliary power, however, was an important item. The superior results secured with the pneumatic distributor led to decision that it was superior for use at sea, and, consequently, this type of distributor was installed on the S. S. *Mercer* prior to its fifth transatlantic voyage.

Problems were continually created and often solved aboard the *Mercer* as a floating pulverized coal laboratory at sea, while the Fuel Oil Testing Plant was also working out solutions to many of the same problems, furthering development, and testing new equipment ashore.

The successful operation of the pneumatic distributor aboard the *Mercer* allowed part of the Plant's attention to be turned toward completion of the development test of the Fuller-Lehigh Pulverized Coal Installation itself. That company—during the period of distributor development—had been perfecting a new design of pulverizer known as the Table Mill, which is a vertical reduction machine in which three large balls are used instead of the large number of small balls employed in the Bonnot Mill in common with the Kennedy-Van Saun Pulverizer—both of which were described in the previous article. It was asserted by the manufacturer that use of this newly designed mill would entail considerable saving of weight and allow large economies in power consumption to be effected without sacrificing fineness of grind. Substitution of the Fuller Table Mill for the Fuller-Bonnot Mill

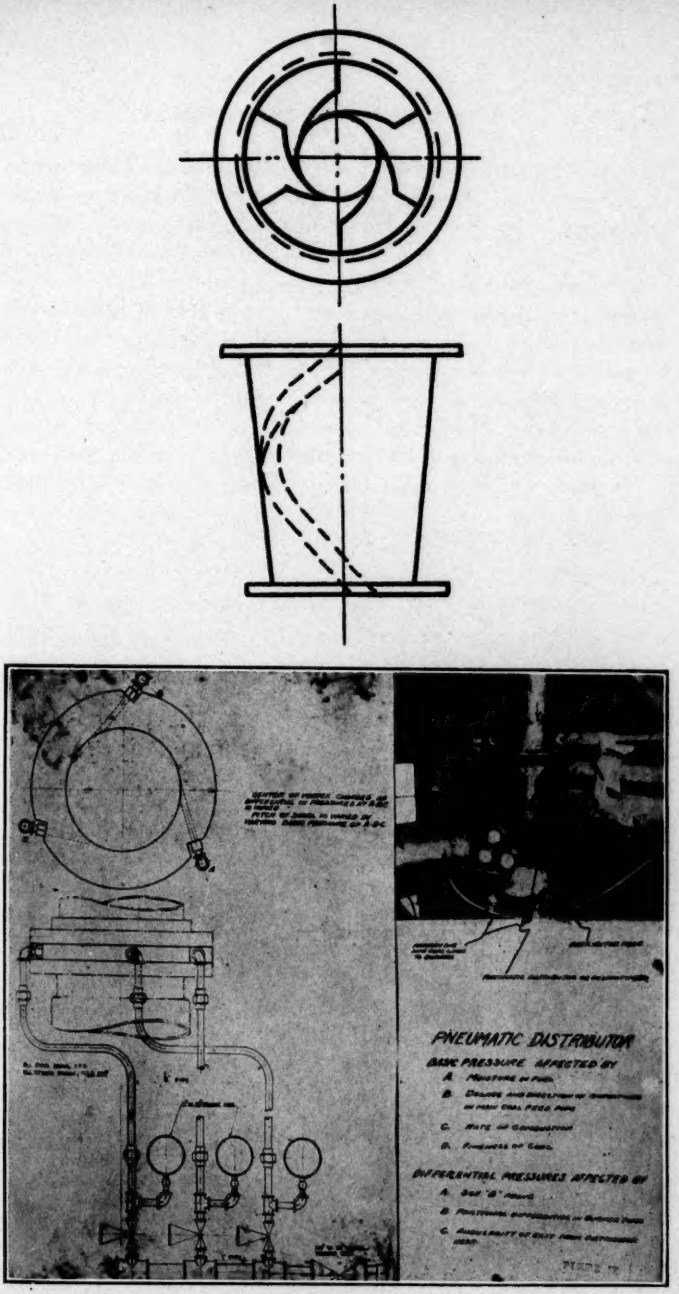

PNEUMATIC DISTRIBUTOR
BASIC PRESSURE AFFECTED BY
DIFFERENTIAL PRESSURES AFFECTED BY

FIG. 1.

as part of the test installation therefore seemed desirable in order to furnish comparison between the two mills, with the idea of employing for final test of this installation the one which seemed more suitable for marine use.

The Todd Engineering and Drydock Corporation, however, had developed a pulverized coal burner to a degree where, during the early summer of 1928, they believed its use would solve many of the problems encountered with other installations. Authority for conducting a test of this equipment was therefore requested by them; the Fuller-Lehigh Company very kindly agreed to allow the Todd test to proceed, postponing the installation of their table mill until the completion of the Todd test.

TODD PULVERIZED COAL BURNERS.

The installation to be tested consisted of three Todd Pulverized Coal Burners and coal pulverizers — a unit* system in which each burner has a separate mill constructed integrally with it. The pulverizers are of the high speed centrifugal type — as differentiated from the low speed ball mill type pulverizer used on two preceding pulverized coal tests. The burners are of the turbulent type; their function is the production and maintenance of efficient combustion of pulverized coal, or if necessary or desirable, fuel oil.

The burner is composed of four principal parts:

(1) Pulverizing unit including the mill itself, mill feeder, and driving motor for the whole burner. The pulverizer elements of the mills were designed and constructed by the Erie City Iron Works of Erie, Pa.

(2) The coal injector — which passes coal into the furnace with the designed purpose of introducing it in such a way as to permit ready intermixture of the air with the diffused coal particles.

(3) The air register — which has the function of introducing "secondary" air into the burner throat to procure intermixture of air with the diffused coal (or oil spray if the fuel oil atomizer be in use).

* A "Unit System" is one in which the coal is pulverized continuously immediately before passing to the burners, thus avoiding the dangers of the "bin system" in which the coal is pulverized, conveyed to storage, and drawn upon as needed. The "bin system" is, at this time, impractical for ship installations because of difficulties, encountered in maintaining bunkers gas-tight as well as dust tight. In order to avoid any possibility of spontaneous combustion it would be necessary to keep some inert gas— CO_2 for example in such bunkers.

(4) A mechanical atomizer of the Todd Type is used for atomizing fuel oil. This is ordinarily used solely for lighting off the coal, but may be operated, with the air register, as a fuel oil burner. The atomizer may be easily withdrawn from the register after use in lighting off coal; a detachable coupling at the fireroom end of the atomizer barrel allows it to be entirely removed or replaced at will.

One of the three burners as originally assembled for test is shown in Figure 2; Figure 3 shows essential parts of the pulverizer, while Figure 4 is a view of the pulverizer partially assembled.

The driving unit for the pulverizer and burner is a fifteen horsepower, 220 volt, 3 phase, alternating current motor, which, of course, runs at a constant speed when in operation. This speed is 3600 R.P.M. The shaft of this motor is a hollow tube which forms the shaft of the pulverizer and fan. The fuel oil atomizer passes through a stationary sleeve in the center of the hollow shaft.

The pulverizer consists of two stages with separate rotors, each having a set of hammers, the first stage makes use of four hammers, the second six. The fan, which is located beyond the two mill stages, draws the pulverized coal from the pulverizer and discharges it through a suitable diffusing device into the furnace.

In operation coal reaching the feeder — see Figure 8 — passes downward by gravity into the first stage of the mill through the entrance shown as mark 4 in Figure 4. After being partially pulverized by the first stage hammers — see mark 2, Figure 3, the coal passes through the center of the second stage liner, mark 3, Figure 3, to the second stage rotor mark 4, Figure 3, where pulverization is completed. After having been pulverized, the coal passes through the center of the diaphragm — mark 5, Figure 3, into the fan — mark 6, Figure 3. As previously stated, the fan having pulled the coal through the mill, discharges it through a diffuser into the furnace. The diffuser originally furnished with the burner is shown as mark 5, Figure 2. The air to be mixed with the pulverized coal passes through an air control register — the burner, as originally furnished, with the register, is shown as mark 4 in Figure 2.

A door shown as mark 1 in Figure 2, is provided in the first stage for cleaning out any particles of foreign material, such as tramp iron, which may enter the mill. Such material is carried to the rear side of this door and can be easily removed.

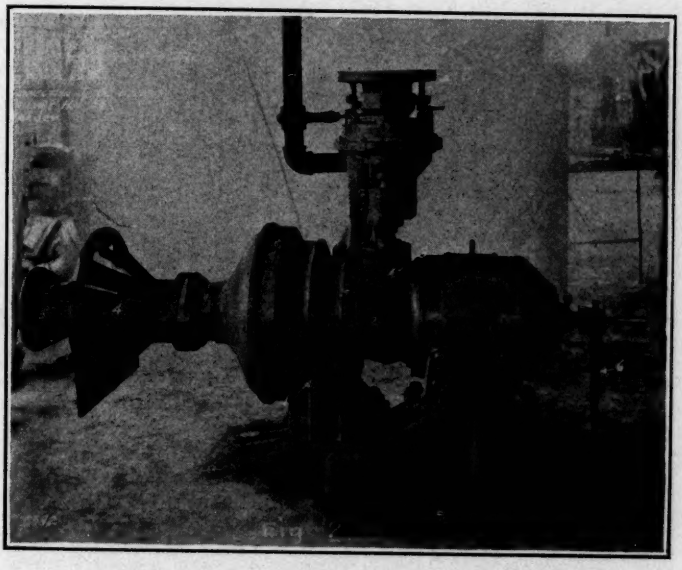


FIG. 2.

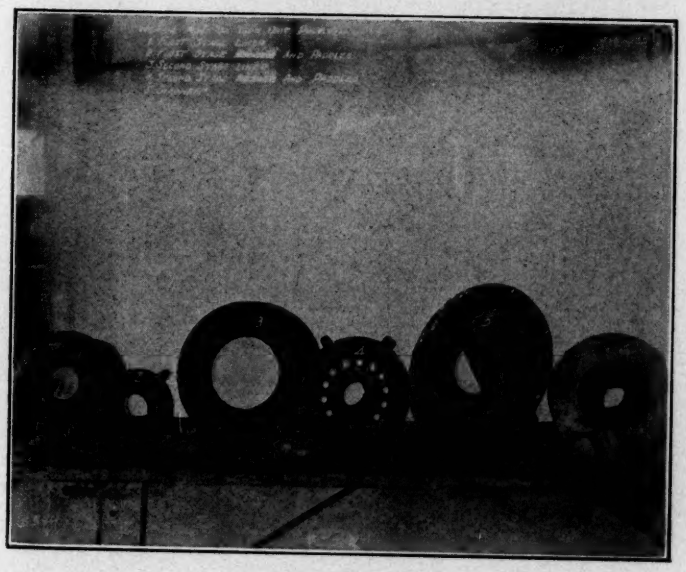


FIG. 3.

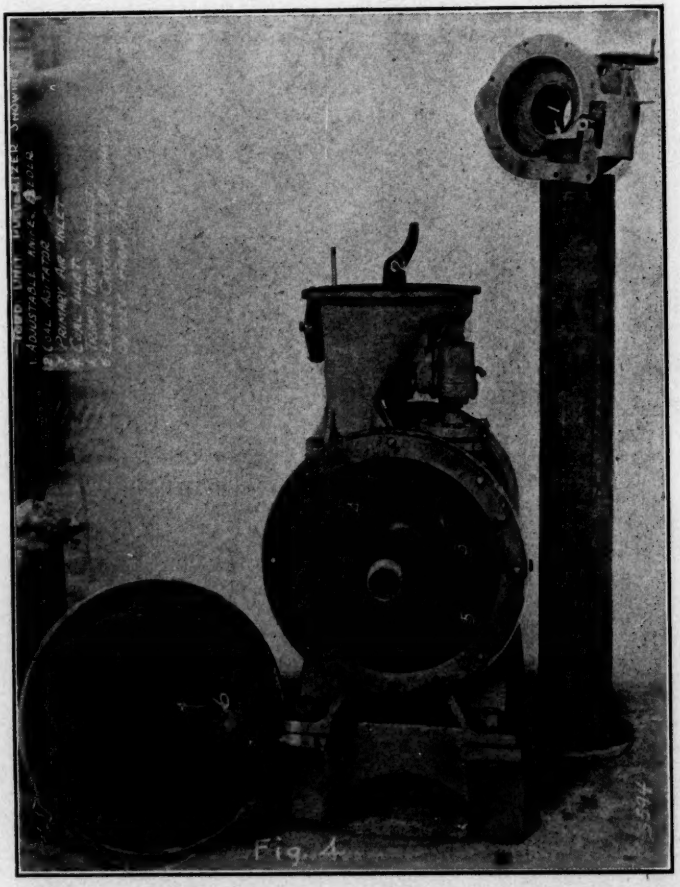


FIG. 4.

THE BOILER.

The Fuel Oil Testing Plant Single Ended Scotch Marine Boiler, upon which all previous pulverized coal tests had been made, was again used for runs with the Todd Pulverized Coal Burners. This boiler was installed some years ago for use in fuel oil burner tests for the Shipping Board, and in instructing the engineering personnel of the Fleet Corporation in the burning of fuel oil. A sketch of this boiler with the air heater and double front as installed is shown in Figure 5.

The principal particulars of this boiler as used on the present test were exactly the same as they were during the two previous pulverized coal tests, but are repeated herewith for additional clarity:

Total boiler water heating surface, square feet	2555
Inside diameter, feet and inches	14–9
Overall length of boiler, inches	132
Distance between tube sheets, inches	84 14/32
Working pressure, pounds per square inch, gauge	200
Number of furnaces	3
Diameter of furnaces at bottom of corrugations	43
Length of furnaces, feet	7
Furnace heating surface total, square feet	270
Number of ordinary tubes	270
Number of stay tubes	94
Total number of tubes	364
Inside diameter of ordinary tubes, inches	2¾
Inside diameter of stay tubes, inches	2⅜
Total number of tubes from each wing combustion chamber	122
Total number of tubes from central combustion chamber	120
Tube heating surface, square feet	2015
Inside distance back to front of combustion chambers, inches	34
Total combustion chamber heating surface, square feet	270
Volume of each furnace and its combustion chamber without any brickwork, cubic feet	161
Total volume of the three furnaces and combustion chambers without any brickwork, cubic feet	482.9
Total volume of the three furnaces and combustion chambers with brickwork as installed on test	427.6
Number of gas tubes in air heater over each wing furnace	163
Number of gas tubes in air heater over center furnace	42
Total number of tubes in air heater	368
O. D. of air heater gas tubes, inches	3
Length between sheets of air heater, inches	25¼

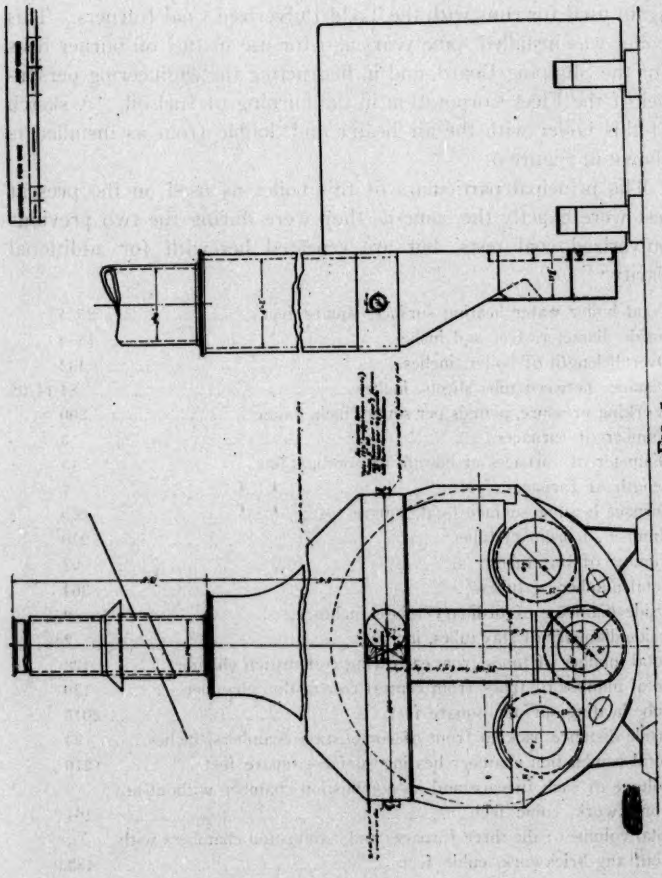


FIG. 5.

Air to the air heater (which passed from it into the double front and thence to the air control registers) was supplied by a centrifugal fan, 63 inches in diameter, driven by a 4½-inch stroke reciprocating engine.

All boiler tubes were equipped with retarders at the ends contiguous to the combustion chambers. The boiler had individual combustion chambers for each of the three furnaces. The rear sheets of the combustion chambers were water cooled.

INSTALLATION.

The registers of the burners were bolted to the furnace front of each furnace and the subject burners installed, one per furnace. The pulverized coal nipple (from the fan housing to the register and coal diffuser) being bolted to air adaptor door which was installed on the front plates of each of the Howden fronts — see Figure 6, which shows registers as installed just prior to putting on the adaptor doors. Structural steel supports were used for each burner. The complete installation as installed, but after certain modifications had been made prior to official runs, is shown in Figure 7.

Coal for use during test was stored on a platform outside the fireroom. It was lifted from this platform into an outside hopper (coal surface about ten feet above the platform) by an endless chain bucket conveyor. From this outside hopper, the coal fell by gravity into a sheet metal bin on the platform of a Howe scale. The coal after weighing, was allowed to fall through a chute which carried it by gravity to the foot of a vertical bucket conveyor, which lifted it from the chute about eighteen feet and discharged it into a screw conveyor running parallel with the front of the boiler uptake. This conveyor discharged the coal through lever operated doors into three "ready" bins, from which the coal passed downward through 8-inch sheet iron pipes into three constant level funnels immediately above the burner feeders. These constant level funnels were supported above the feeders by threaded bolts; the height of the funnels could be easily controlled by adjusting the supporting nuts. As the height of the funnels controlled the flow of coal to the feeders as well as the flow from the conducting lines to the funnels, the feeders were allowed to remain wide open at

all times; rate flow of coal to the burners being controlled by the adjustment of the funnels.

During all runs, coal was passed to the "ready" bins alternately — in other words, first to No. 1 bin, then to No. 2 bin, and last to No. 3 bin, in order to check the coal rate per burner as well as the total coal rate — thus checking the distribution of coal to the individual furnaces. Elevation of the burner installation with "ready" bins and other details, is shown in Figure 8.

Oil used for lighting off was weighed only to determine the amount consumed for this purpose.

Feed water to the boiler was weighed in two water weighing tanks, fed alternately from the Plant water main under approximately sixty pounds pressure — water weighing tanks discharged alternately into constant level water tank from which suction was taken by a Warren Steam Reciprocating Pump and forced via a Reilly Multicoil feed water heater through the feed line to the boiler stops and check valves.

The steam pressure on the boiler, and the main steam line from the boiler, were maintained at constant pressure by a hand controlled bleeder valve through which the steam exhausted directly to the atmosphere. A steam jet installed at the base of the stack was used to increase the draft when simulating various stack heights. A throttling calorimeter was used to ascertain the quality of steam.

Description of other test apparatus, pumps, indicators, and recorders, used is not included in the present article; it suffices to say that the instruments installed were carefully calibrated and adequate for the ascertainment of all pertinent data.

COAL USED DURING TEST.

Six varieties of coal were used during the Todd Test. Three of these were Pennsylvania semi-bituminous coal, and included:

(1) Keystone Coal & Coke Company's Greensburg Crushed Coal from their Crow's Nest Mine, Hempfield, Westmoreland County, Pennsylvania (Greensburg Basin, Pittsburgh seam, semi-bituminous).

(2) White Star Coal Co., subsidiary of the Buffalo Rochester and Pittsburgh Coal & Iron Co., Luzerne County semi-bituminous.

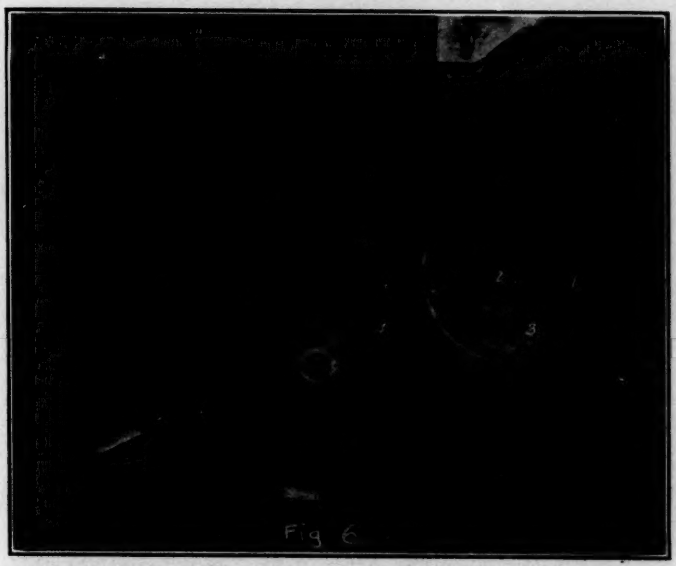


FIG. 6.

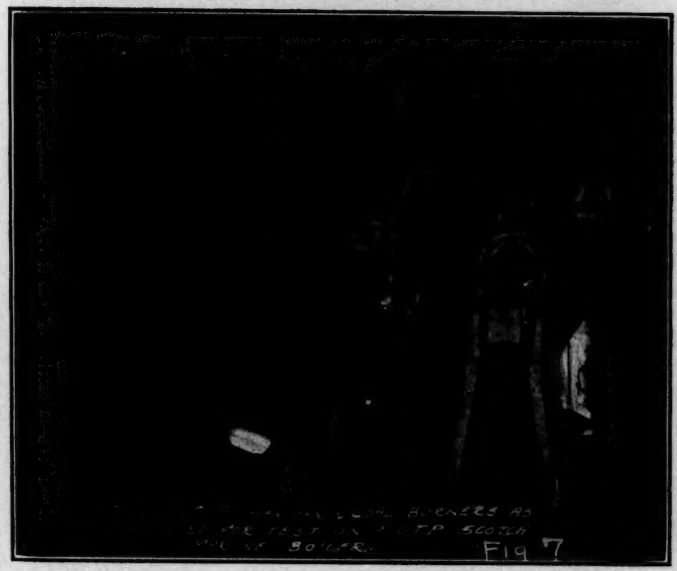


FIG. 7.

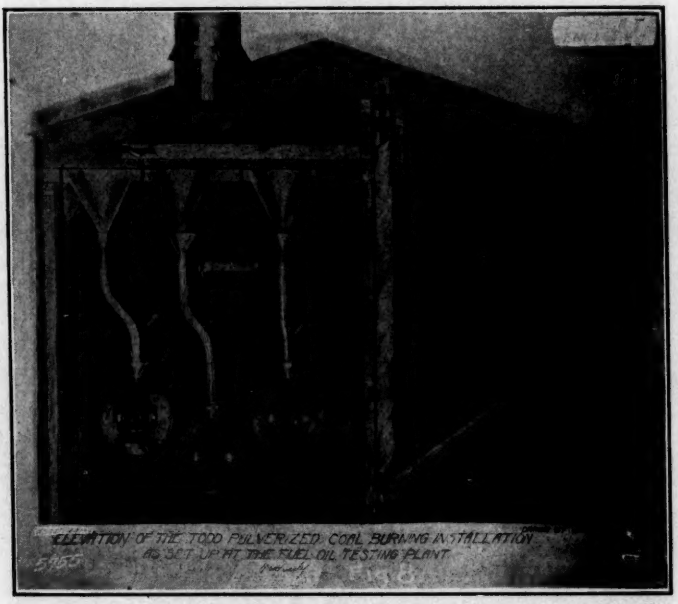

ELEVATION OF THE TODD PULVERIZED COAL BURNING INSTALLATION
AS SET UP AT THE FUEL OIL TESTING PLANT

FIG. 8.

This coal was delivered crushed, and was called "pulverized coal size."

(3) Morrisdale Coal Co., Morrisdale Shaft No. 1, Clearfield District, Clearfield County, Pa. (Miller "B" seam semi-bituminous). This was slack coal.

The other three coals employed were:

(4) Brazilian bituminous, mined at Sao Jeronymo in the State of Rio Grande do Sul by the Companhia Mina Sao Jeronymo. It was furnished by the Cia de Navegagao Lloyd Brasileiro, Rio de Janeiro. This coal was of low heat value and high ash content. It was received in the form of large, hard lustreless black lumps stowed in 200-pound gunny sacks.

(5) Westphalian semi-bituminous coal mined in the State of Westphalia of the same grade as that purchased by the Shipping Board in Rotterdam for use aboard the S. S. *Mercer*. The test supply being donated by Coal Contractor.

(6) A very small amount of semi-anthracite mined in the district of the Philippine Islands — Island of Mindanao — by the Philippine National Coal Company was used for a very short interval on a special run in order to determine the grind obtainable by its use in connection with this mill.

PULVERIZED COAL SAMPLES.

Pulverized coal samples were taken at least once during each operating condition, and during each eight-hour period of the seventy-two-hour simulative service run (described below).

The 6-inch pipes connecting the burner fan discharges with the burner openings proper, were drilled at the top and bottom on the vertical center line for ½-inch pipe tap connections. Two ⅜-inch pieces of brass pipe 5 inches long half cut away for a distance of ¾ inch at one end, and projecting a third of the distance across the coal stream, were placed through these sampling holes. The ¾-inch cut away ends faced the coal stream in the burner. The outer ends of these brass nipples were fitted with bushings to which aspirators were attached. When sampling, the aspirators were operated by compressed air which sucked coal from the sampling tubes and blew it into a vacuum cleaner bag. Coal was drawn from

all sampling tubes for equal periods. The time of sampling was sufficient to allow approximately five pounds to be collected from each tube.

COAL SAMPLES FOR ANALYSIS.

Coal samples for analysis were obtained by taking a 300 cubic centimeter scoopful from the coal scale hopper prior to the passage of the coal to the feeder. Suitable composites of these samples were obtained and packed in air-tight cans. Analyses were made by the Bureau of Mines (Department of Commerce) and also by the Fuel Oil Testing Plant — a third check on proximate analysis and calorific values was made by the United States Shipping Board.

TEST PROCEDURE.

The runs of this test may be subdivided into four principal groups:

(1) Preliminaries. About forty runs were conducted, during which data was obtained on the installation as received and certain small mechanical improvements made between runs.

(2) Run conducted for seventy-two hours without shut down or securing coal in order to determine difficulties, if any, encountered during prolonged periods of operation.

(3) Spiral Diffuser Runs — conducted to ascertain whether this diffuser allowed better results to be obtained.

(4) Coal Runs — conducted with spiral diffusers using Westphalian and Brazilian coal, in order to determine the practicability of operation with these coals. A brief run was made with Mindanao coal principally to investigate the fineness of grind obtainable from its use in these mills.

The combustion rates at which the boiler was operated were: 300, 400, 500 and 600 pounds per burner per hour, or 900, 1200, 1600 and 1800 pounds per hour total for the three burners. During the seventy-two-hour run each of these conditions was maintained for sixteen hours — except the 1200 pounds per hour run, which was continued for twenty-four hours to allow for steadying of conditions during the first eight hours of operation.

During all runs the smoke was maintained at a minimum. Steam pressure was maintained constant at 180 pounds per square inch gauge at all times.

For the seventy-two-hour run, the personnel of the Fuel Oil Testing Plant was divided into three equal parts, each making up an eight-hour watch. The engineers of the Shipping Board who were present and Messrs. Haines and Keller of the Todd Engineering and Dry Dock Company assisted on this duty. Lieutenant-Commander F. D. Kirtland, U. S. N., of this Plant and Mr. Charles Jack of the Shipping Board were in charge of two of these watches, the remaining one being headed by the author of this article. Commander J. J. Broshek, U. S. N., of the Plant and Mr. C. J. Jefferson, Head of the Fuel Conservation Section of the Shipping Board, acted as supervising and consulting engineers during all runs.

RESULTS.

A thorough analysis of all knowledge gained during this test is impossible in the space of this monograph; the most interesting results accomplished beyond actual efficiencies themselves, which will be considered further on, were:

(1) INCREASED STEADINESS OF OPERATION THROUGH CONSTANCY OF COAL FEED TO THE MILLS.

In the installation as originally set up for test, the hoppers for supplying coal to the feeders were integral with the coal supply lines, the hoppers themselves being furnished with gate valves at their lower extremities to allow regulation of the amount of coal passing to the mill feeders.

Difficulty was experienced in obtaining a steady flow of coal from these hoppers and valves. They were therefore discarded and, after several runs to determine the most suitable dimensions, feed funnels into which the coal supply lines projected downward were installed. Grooves were cut in the revolving plates of the mill feeders to more evenly distribute the flow of coal, and the length of the feeder knives or scrapers was increased. It was found that, after these changes, the rate of feed could be easily varied over the range of rates desired simply by raising or lower-

ing the height of the feeder funnels. The height of coal in these funnels kept it from flowing out of the supply pipes faster than the feed rates desired and also controlled speed of efflux onto the revolving disc of the feeder.

NOTE: The speed of the mill and main drive shaft were constant at 3600 R.P.M., the speed of the pinion and coupling to feeder gears 100 R.P.M., the feeder gears reducing the speed of the revolving table in the ratio of 1 to 18.52 turns of the coupling.

(2) ALTERATIONS TO COAL AND AIR DIFFUSING DEVICES.

The burners were originally furnished with slotted cones for coal diffusion — refer to mark 5, Figure 2. These cones, however, did not diffuse the pulverized coal to a sufficient degree to produce the best results; they also became burned and eroded after very short periods of use. The cones were, therefore, removed and short diffusing fins, shown in Figure 9, installed in the burner throats. At the same time, the air admission area of the air control registers was restricted to cause greater velocity through the vanes, in the hope of securing better air penetration of the diffused coal. These alterations made such an improvement in the intimacy of pulverized coal and air intermixture that it was decided to make a seventy-two-hour run with the burners thus modified. Results from that period of operation, however, demonstrated that intermixture was still rather imperfect, particularly at the rates of 500 and 600 pounds per burner per hour. Particles of pulverized coal were not only insufficiently disseminated, but the force of air entering the air registers cast the burning particles so far into the furnaces that much of the combustion occurred in the rear of the combustion chambers — against the rear sheets.

These imperfections in diffusion and intermixture led to the designing of an angled blade, or spiral diffuser, made up to impart wider dissemination and spiral flow to the coal particles. It was patterned, in a general way, after the spiral distributor described in the opening paragraphs of this article. While these diffusers were being installed in the burner throats, the air admission area of the registers was increased. The fin diffuser and the spiral diffuser in the burner throats are shown in Figure 9. Diffusion of pulverized coal and its intermixture with the air were greatly

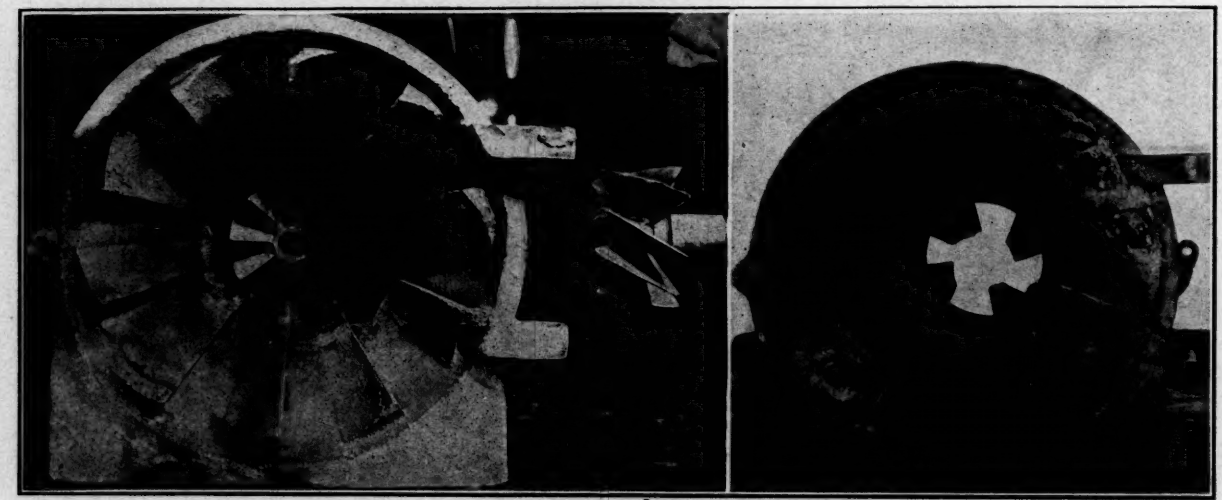


FIG. 9.

improved by use of the spiral diffusers. This was best demonstrated by results which will be considered further in this article.

COAL GRIND.

The companies supplying the coal used have been previously noted; analyses of the various coals during runs of test were:

	Pennsylvania Coal			Foreign Coal		
	Keystone	White Star	Morrisdale	Brazilian	Westphalian	Mindanao
Moisture	2.7	6.7	3.1	8.5	2.8	2.8
Volatile Matter	29.6	25.4	24.0	26.6	31.9	10.9
Fixed Carbons	56.5	60.7	63.2	34.5	55.2	72.6
Ash	11.2	7.2	9.7	30.4	10.1	13.7
Totals	100.0	100.0	100.0	100.0	100.0	100.0
Fusion Point of Ash, degrees F	2500	2490	2490	2620	2320	2920
H	5.1	5.5	5.1	4.3	5.2	4.0
C	73.5	75.0	76.8	47.2	73.1	74.4
N	1.5	1.4	1.3	0.8	1.4	1.9
O	6.9	9.4	5.3	15.8	8.5	5.4
S	1.8	1.5	1.8	1.5	1.7	0.6
Ash	11.2	7.2	9.7	30.4	10.1	13.7
Totals	100.0	100.0	1.00	100.0	100.0	100.0
B.T.U./lb.	13230	13250	13560	8200	12980	12660

All the varieties of coal used were practically free from foreign matter — such as tramp iron. The friability of the coal is naturally dependent upon the type of coal used — its hardness, cleavage and fracture.

The required power input to the pulverizers varied with the friability of the coal used, the amount of moisture intrained in it, and the rate at which it was fed. The degree of pulverization or fineness of grind was also a variable dependent upon the same factors. At any given rate, therefore, the fineness of grind decreased as less friable and harder coal was used; for any given coal the fineness of grind decreased as the rate of feeding was increased.

The fineness of grind obtained at the 400-pound per burner per hour rate using the six different varieties of coal is shown in Figures 10 and 11. It will be seen at once from the circular percentage charts of Figure 10 that the grinds procured when using Pennsylvania coals — namely, Keystone, White Star and Morrisdale — were all very similar. The grind with the less friable Westphalian coal was inferior to that with the Pennsylvania, while the even less friable and harder Brazilian and Mindanao types gave still coarser pulverization. The grind of these six coals is also plotted on the triangular percentage chart, Figure 11. With the percentage scales arranged as they are in that figure, the nearer the points to the lower left-hand corner, the finer the grind, as the percentage of material through 300-mesh per square inch is at its maximum with percentages of coarses and intermediates at a minimum at the lower left-hand extremity of the chart. It will be noted also from this figure that grinds gained with the Pennsylvania coals were all similar to each other, while the Westphalian, Brazilian and Mindanao coals gave coarser grinds in that order.

The grinds obtained with these six coals were briefly as follows:

	Keystone	White Star	Morrisdale	Brazilian	Westphalian	Mindanao
100 M. or coarser	4.4	1.7	1.4	14.6	11.2	11.6
300 M. or finer....	90.8	88.9	90.2	72.0	76.8	66.4

The influence of rate on grind with a given coal is shown in Figures 12 and 13. Figure 12 shows circular percentage charts of the grinds obtained with White Star coal at the rates of 250, 350, 400, 450, 500 and 550 pounds per burner per hour; Figure 13 shows the same data plotted on a triangular percentage chart. It will be noted from these figures that the fineness of grind decreased as the rate increased. Figure 13 also shows that, with this particular coal, as the percentage of very finely pulverized coal decreased with increased rate, the percentage of coarses increased slowly, while the percentage of intermediate fineness increased rapidly.

It will be immediately apparent from these charts that exceptional fineness of grind was obtained with the Todd pulverizers.

CIRCULAR PERCENTAGE CHARTS SHOWING GRIND OBTAINED WITH SIX KINDS OF COAL AT 400 LBS. PER BURNER PER HOUR.

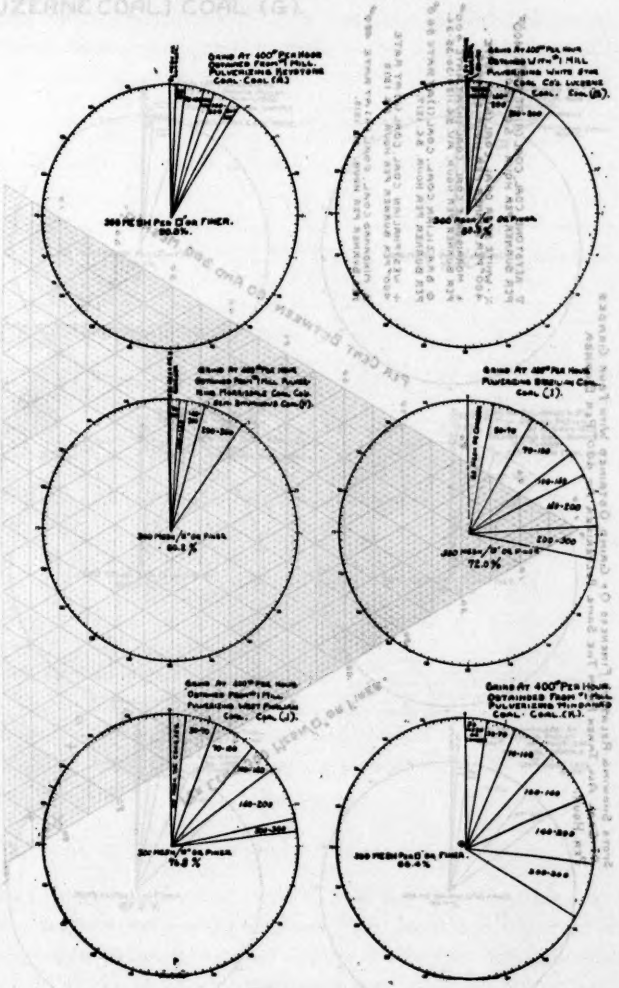


FIG. 10.

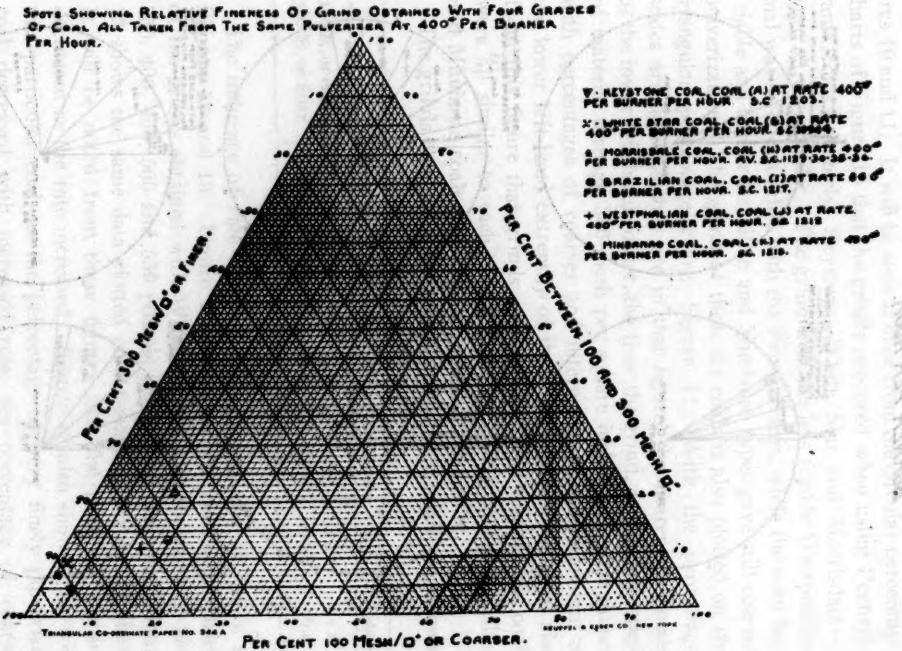

Spots Showing Relative Fineness Of Grind Obtained With Four Grades Of Coal All Taken From The Same Pulverizer At 400# Per Burner Per Hour.
V - Keystone Coal, Coal (A) at rate 400# per burner per hour S.C. 1203.
X - White Star Coal, Coal (G) at rate 400# per burner per hour S.C. 1214.
A Morrisdale Coal, Coal (H) at rate 400# per burner per hour. Av. S.C. 1159-30-30-30.
B Brazilian Coal, Coal (I) at rate 86# per burner per hour. S.C. 1217.
+ Westphalian Coal, Coal (J) at rate 400# per burner per hour. S.C. 1218
D Mindanao Coal, Coal (K) at rate 400# per burner per hour. S.C. 1219.
Per Cent 200 Mesh/□" or Finer.
Per Cent Between 100 And 200 Mesh/□"
Per Cent 100 Mesh/□" or Coarser.
Triangular Co-ordinate Paper No. 344 A
Keuffel & Esser Co. New York

FIG. 11.

CIRCULAR PERCENTAGE CHARTS SHOWING GRIND OBTAINED AT VARIOUS RATES DURING PRELIMINARY RUNS USING WHITE STAR (LUZERNE COAL) COAL (G).

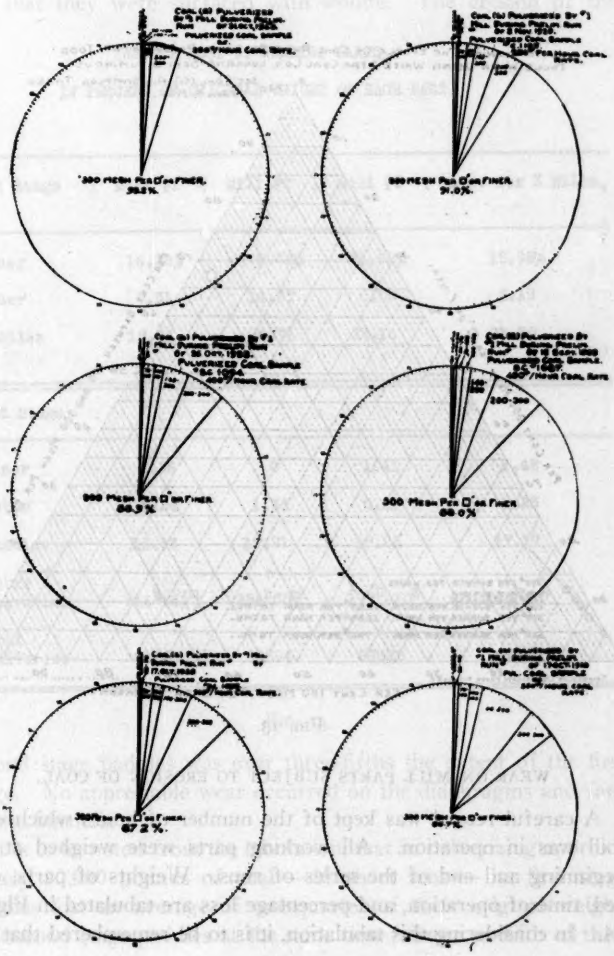


FIG. 12.

At none of the four rates (300, 400, 500, 600 pounds per burner per hour) which were run with all the Pennsylvania coals, did the percentage of powdered coal of 300 mesh per square inch or finer fall below 85 per cent, nor did the percentage of material 100 mesh per square inch or coarser at any time exceed 7 per cent.

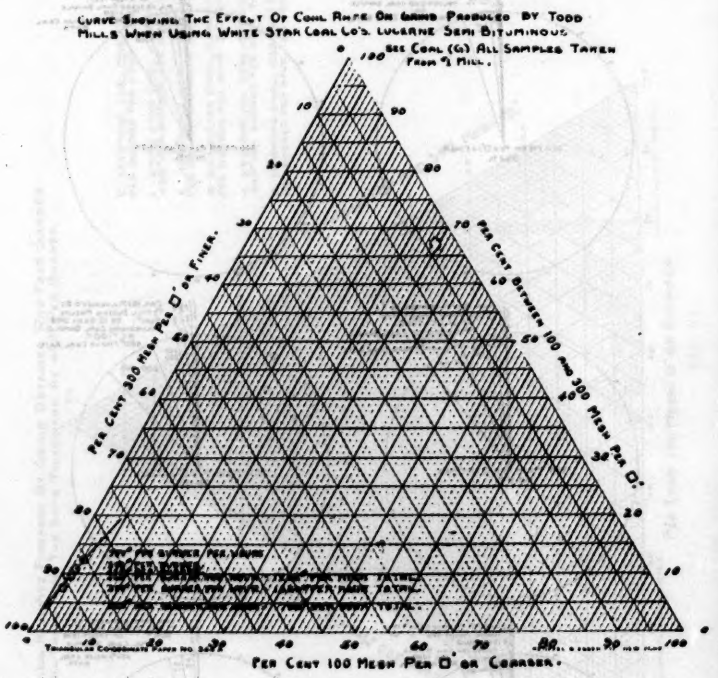

Curve Showing The Effect Of Coal Rate On Grind Produced By Todd Mills When Using White Star Coal Co's. Lucerne Semi Bituminous
See Coal (G) All Samples Taken From 4 Mill.
Per Cent 300 Mesh Per □" or Finer.
Per Cent Between 100 and 300 Mesh Per □"
Per Cent 100 Mesh Per □" or Coarser.

FIG. 13.

WEAR ON MILL PARTS SUBJECT TO EROSION OF COAL.

A careful record was kept of the number of hours which each mill was in operation. All working parts were weighed at the beginning and end of the series of runs. Weights of parts, coal fed, time of operation, and percentage loss are tabulated in Figure 14. In considering this tabulation, it is to be remembered that the

first stage hammers of the mills were soft steel, while those of the second stage were surfaced with stellite ¼ inch thick. All liners were of soft steel.

The greatest wear occurred on the first stage paddles; the second stage paddles received the next most severe wear, in spite of the fact that they were surfaced with stellite. The erosion of the

WEIGHT LOST BY WEARING PARTS
IN PERCENT OF ORIGINAL WEIGHT OF EACH PART

1st Stage	Mill #1	Mill #2	Mill #3	Avg. for 3 Mills.
Liner	16.28%	9.75%	21.82%	15.95%
Rotor	0.51	14.87	0.50	5.19
Paddles	18.64	34.71	23.16	25.50
2nd Stage				
Liner	2.86	0	1.41	1.42
Rotor	-1.24	1.65	0.81	1.25
Paddles	16.04	17.21	18.25	17.17
Hours Operation	216h-54m	254h50m	197h-10m	222h31m
Coal Pulverize	87527	99594	80322	89148

FIG. 14.

second stage paddles was over three-fifths the extent of the first stage. No appreciable wear occurred on the diaphragms and fans of the burners.

It is to be noted from the tabulation, that in pulverizing approximately 90,000 pounds of coal — 45 short tons — the first stage paddles lost an average of one-quarter of their original weight, the second stage paddles at the same time losing one-fifth of their

original weight. The losses of approximately one-eighth of the weight of the first stage liners was not to be overlooked. The Todd Engineering and Drydock Company are now conducting experiments with other material which, it is believed, will greatly diminish the wear on these parts.

POWER CONSUMPTION OF THE BURNERS.

The power consumption of the burners was taken by means of a recording Esterline Angus Graphic Watt Hour Meter which logged current input continuously while the installation was in operation. The instrument was so installed that the input to each mill was taken for one-hour periods alternately. From the original of this chart, the average line per burner per rate was obtained by means of a planimeter. The average line of reading thus obtained was then multiplied by a factor of twenty, which, on that instrument and chart, converts the line of reading to kilowatts. The power consumption, in kilowatts, was thus obtained for each individual burner (including pulverizer, primary air fan and feeder) for each coal rate run. The average power consumption of the three machines at each rate was then averaged in order to obtain a mean power consumption for any given rate. Curve of the average power data compiled by the Fuel Conservation Section of the Shipping Board are shown in Figure 15. It is not to be overlooked that the power consumption here shown includes that to the mill, primary air fan, and feeder of each burner.

OPERATION WITH THE THREE PENNSYLVANIA COALS.

With the three varieties of Pennsylvania coal used, no difficulties of operation were experienced. There was very little slagging in evidence in the furnaces and there was no coking on the furnace opening rings. The flames were good, and, as the fusion point of the ash was high, no difficulty was experienced in carrying high CO_2.

OPERATION WITH WESTPHALIAN COAL.

With Westphalian Coal some difficulty arose from coking of the coal on the rings because of the high volatile content, but high CO_2 and good furnace conditions could be maintained without difficulty.

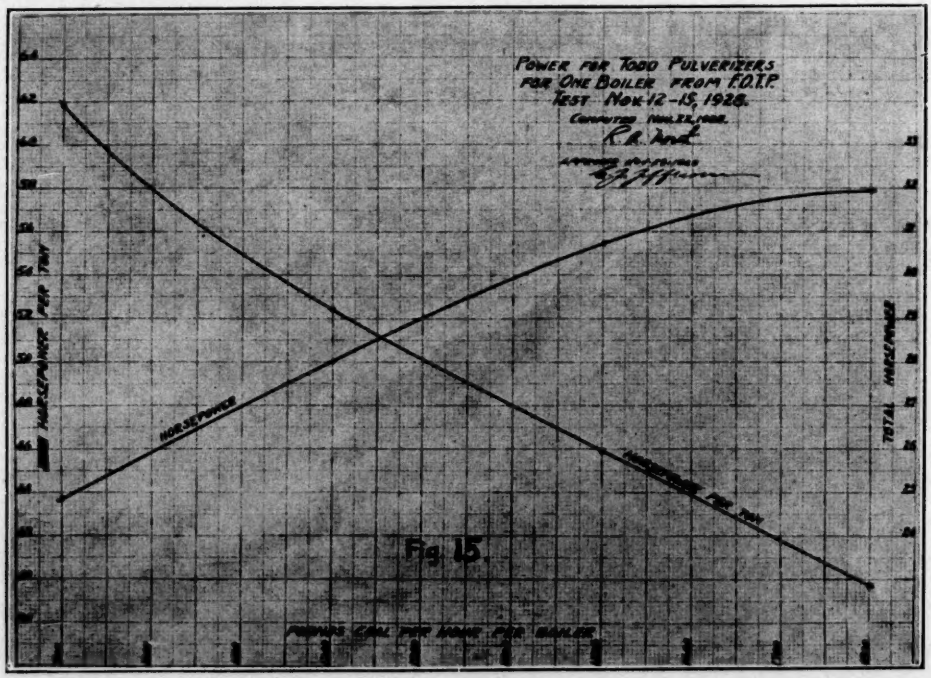

POWER FOR TODD PULVERIZERS
FOR ONE BOILER FROM F.O.T.P.
TEST NOV. 12-15, 1928.
COMPUTED NOV. 15, 1928.
R.R. Ford
APPROVED FOR FILING
HORSEPOWER
HORSEPOWER PER TON
TOTAL HORSEPOWER
CONSUMPTION PER TON
POUNDS COAL PER HOUR PER BOILER
Fig. 15.

FIG. 15.

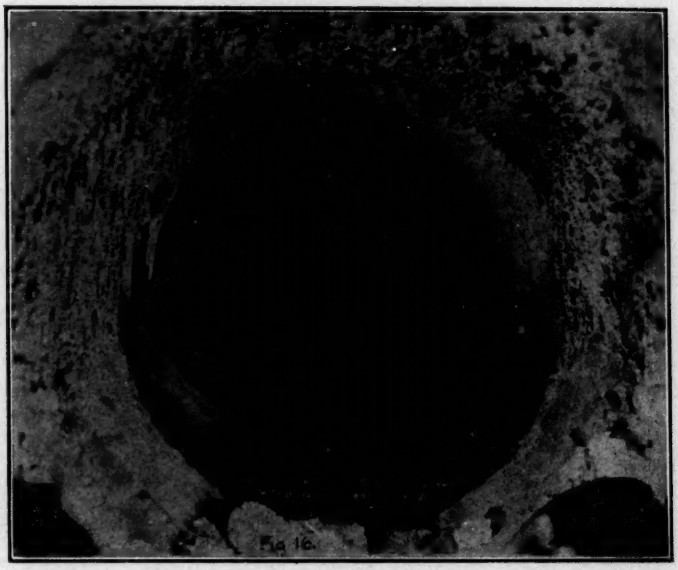


FIG. 16.

OPERATION WITH BRAZILIAN COAL.

The Brazilian Coal burned well in spite of the exceedingly high ash content, very low calorific value and relative coarseness of grind because of extreme hardness. The ease and steadiness of combustion was probably due to the high percentage of oxygen present in this coal.

A large amount of ash naturally was deposited in the furnaces during operation with this coal; this was not only due to the high percentage of ash present in it; but also to the relatively low velocity of gases through the furnaces. Figure 16 shows the appearance of No. 1 furnace after eight hours operation at 360 pounds per burner per hour with this coal.

It is considered that successful operation with this coal constituted one of the outstanding features of the test. Operation when using this coal with Todd Burners was not only possible but entirely feasible, and if means were provided either for blowing out the furnaces by means of specially modified soot blowers or raking the residue from the furnace through suitable doors constructed in the rear sheets, it is believed that steady and continued steaming could be easily accomplished at, of course, reduced boiler rating.

OPERATION WITH MINDANAO COAL.

Steady operation with Mindanao coal proved impossible with the small amount of this coal on hand. The grind obtained was considerably coarser than that procured with Brazilian coal — refer to Figure 11. After repeated attempts were made to light this coal off, it could not be made to burn for more than two or three minutes. While the flame burned, it was of the dull red color characteristic of some anthracite coals. The difficulty in burning was probably due to the very low percentage of volatile material and hydrogen present, as well as to the hardness of the coal — which precluded a fine grind from being obtained. Had a larger quantity of this coal been available at the time of this run it might have been possible to devise better conditions for its combustion.

SIZE OF COAL FOR PULVERIZATION.

Coal used in the Todd mills — regardless of grade or quality — must be slack, sized, or broken in a crusher prior to passing to the

mill feeders. The mills and feeders are capable of accommodating pieces up to ¾ inch diameter at high rates per burner; for rates below 400 the coal particles should not be larger than ½ inch diameter. Larger lumps lead to erratic feed or to choking of the feeders, these conditions lead to unsteady grind and fluctuations in combustion, in some instances causing the flames to momentarily die out.

ANALYSES OF SMOKE BOX RESIDUES.

Analyses and calorific value determinations were made on samples consisting of various mesh particles taken from the smoke boxes at the end of the 72-hour run. Representative samples of the thoroughly mixed deposited material were carefully screened according to the usual method of obtaining fineness of grind with pulverized coal. About fifty grains of each grade of fineness, viz.: coarser than 50 mesh, 50-70, 70-100, 100-140, 140-299, 200-300,

TABULATION SHOWING ANALYSES OF SOOT OF VARIOUS FINENESSES TAKEN FROM SMOKE BOXES #1 AND #2, AFTER 72 HOUR SIMULATIVE SERVICE RUN.

	SMOKE BOX #1					
	Moisture %	Volatile %	Carbon %	Ash %	Sulphur %	B.T.U.'s
coarser than 50 m.	0.30	3.00	41.85	55.15	0.40	6475
50 - 70	0.40	3.00	58.85	38.15	0.90	9035
70 - 100	0.20	3.00	61.80	35.20	0.35	8875
100 - 140	0.40	3.00	62.60	34.40	1.00	9460
140 - 200	0.40	3.00	65.90	31.10	0.55	9740
200 - 300	0.50	3.00	68.85	28.15	0.90	10050
300 or Finer	0.30	4.00	56.50	39.50	0.41	8375
	SMOKE BOX #2					
coarser than 50 m.	0.30	6.40	20.35	73.25	0.38	3010
50 - 70	0.40	3.00	52.30	44.70	0.35	7930
70 - 100	0.40	3.00	51.80	45.20	0.36	7665
100 - 140	0.60	3.00	59.75	38.25	0.77	9100
140 - 200	0.40	3.00	63.90	33.10	0.60	9300
200 - 300	0.60	3.00	66.80	30.20	0.60	9970
300 or Finer	0.30	4.00	54.90	41.10	0.35	8050

FIG. 17.

ABRIDGED DATA SHEET

1	THE FUELS	
2	BOILER HEAT HEATING SURFACE - Sq. Ft.	
4	GROSS FURNACE VOLUME - Cu. Ft.	
5	FURNACE HEAT. VOL. CU. FT. - AFTER INSTALLATION OF POWER BOILER UNIT	
20	FUEL OIL BURNING PLANT NO. OF COAL BARGES	
21	FIXED CARBON	
22	VOLATILE MATTER	
23	MOISTURE	
24	ASH	
25	SULFUR	
26	NITROGEN - MOISTURE FREE BASIS	
27	CARBON (C)	
28	HYDROGEN (H)	
29	SULFUR (S)	
30	NITROGEN (N)	
31	OXYGEN (O)	
32	ASH	
33	TOTAL (AS FIRED)	
34	CALORIFIC VALUE B.T.U.	
70	RETAINED ON #40 SIEVE	
71	RETAINED ON #60 SIEVE, PASSING #40 SIEVE	
72	RETAINED ON #70 SIEVE, PASSING #60 SIEVE	
73	RETAINED ON #100 SIEVE, PASSING #70 SIEVE	
74	RETAINED ON #140 SIEVE, PASSING #100 SIEVE	
75	RETAINED ON #200 SIEVE, PASSING #140 SIEVE	
76	RETAINED ON #270 SIEVE, PASSING #200 SIEVE	
77	PASSING #270 SIEVE	
90	CARBON DIOXIDE $\%\ CO_2$	
91	OXYGEN $\%\ O_2$	
92	CARBON MONOXIDE $\%\ CO$	
93	NITROGEN $\%\ N_2$	
94	SMOKE - RINGELMAN SCALE	
110	STEAM PRESS. ON BOILER LBS./Sq.In. ABSOLUTE	
116	DRAFT AT BASE OF STACK IN.	
122	F.D.F. DISCHARGE AIR HEATING SURFACE $°F$	
130	I.D.F. PREHEAT AIR HEATING SURFACE $°F$	
144	AIR HEAT. GAS1 LEAVING HEATER $°F$	
147	AIR HEAT. GAS2 LEAVE 2 HEATER $°F$	
147	FEED. STEAM CALORIMETER	
148	FEED. TEM. STEAM IN BOILER AT BOILER PRESS.	

*[illegible]

†[illegible]

SHEET SHOWING PERTINENT DATA

ANSWER DATA SHEET SHO
PERTINENT DATA

165	EFF. OF FEED WATER HEATING BOILER [illegible]
169	MILL SPEED - R.P.M.
171	GENERATOR SPEED - R.P.M.
173	RELATION OF RPM - HERTZ.
175	COAL FEED - TOTAL LBS. PR HOUR
177	ANNUAL TOTAL EXP. OF B.T.U.S. LBS.
181	COAL FEED PER HOUR - LBS.
183	COAL FEED PER SQ. FT. B.T.U.S. PER HOUR -
185	COAL FEED PER FURNACE PER HOUR - LBS. ASSUMING EQUAL DISTRIBUTION.
189	COAL FEED PER SQ.FT. OF FURN.VOL. PER HR.
191	B.T.U. FEED PER HR. PER SQ. FT. B.T.U.S.
193	B.T.U. FEED PER HR.PER SQ.FT. OF FURN. VO
195	ANNUAL EVAPORATION OF B.T.U.S. PER HR. - L
200	QUALITY OF STEAM - %
201	ANNUAL EVAPORATION PER HR. CORRECTED FOR
203	FACTOR OF EVAPORATION
205	EQUIVALENT EVAPORATION OF B.T.U.S. PER HR.
214	EQUIV. EVAP. PER SQ.FT. B.T.U.S. PER HR. -
216	EQUIV. EVAP. PER LB. OF COAL AS FIRED - LB
206	BOILER HORSE POWER - HRP
207	FURNACE RATING %
211	EQUIV. EVAP. PER HR. PER DRIVING PAIRED
213	EQUIV. EVAP. PER HR. PER DRIVING [illegible]
217	EQUIV. EVAP. PER HR. PER DRIVING PERMIT A
218	EQUIV. EVAP. PER HR. PER DRIVING [illegible] GRATE AND GRID.
219	EQUIV. EVAP. PER HR. PER DRIVING SECOND. A
220	EQUIV. EVAP. PER HR. PER DRIVING PAIRED FURN
240	% ASSIGN ANY OTHER PROPORTIONAL AMT. REQUIRE FOR COMPLETE COMBUSTION.
250	TOTAL LOSS. THEN LOSSES-ANNUAL HEAT BAL
269	HEAT LOSS CHARGEABLE TO B.T.U.S. & EVAPORA
271	HEAT LOSS CHARGEABLE TO FURNACE & BOILER
273	HEAT LOSS CHARGEABLE TO B.T.U.S. - B.T.U.
275	HEAT ABSORBED BY BOILER PER HR. VOL. & S.F
277	OVERALL EFFICIENCY OF B.T.U.S. FURNACE AND HEATER AND BURNER.
279	EFFICIENCY OF B.T.U.S. - %
279	COMBINED EFFICIENCIES OF FURNACE AND BURN
280	COMBINED EFFICIENCIES OF B.T.U.S. FURNACE, HEATER, AND BURNER, ELIMINATING UNAVOIDA LOSSES - %
281	RATING EXPRESSED BY AIR HEATER - %
283	EFFICIENCY EFFICIENCIES IN B.T.U.S. BURNER BOILER - FURNACE DESIGNED.

*NOTE. USE WITH DRIVING FEED AND S

†NOTE. USED ALONG WITH EQUAL DISTRIBU

ET SHOWING
ORTH

Fig. 12.

and finer than 300 were obtained and analyses, including calorimetric determinations, made on each grade. The results of these determinations are shown in Figure 17.

The amount of moisture present in these samples was approximately constant at about 0.4 per cent; the percentage of volatile matter also did not vary except in the coarsest and finest particles. The principal constituents of the residue were carbon and ash, the heat value varying almost directly with these two variables, in other words, the greater the percentage of carbon, the higher the B.T.U. value; the greater the ash content, naturally the lower the B.T.U. value.

EFFICIENCIES.

A tabulation of some of the principal data obtained at various rates of firing employed during the 72-hour run and during runs with spiral diffusers — including operation with Westphalian and Brazilian coal, is shown in Figure 18; efficiency curves are shown in Figure 19.

None of the results obtained on the preliminary runs (those made prior to the 72-hour run) were completely calculated. Data were logged on all these preliminaries, however, principally to have a complete record of the conditions under which they were made, and in order that the value of the mechanical changes then made could be gauged. Inspection of data taken on runs immediately preceding the 72-hour run, showed them unquestionably superior to those before alterations to supply lines, feeders, diffusers and air registers had been made.

During the seventy-two-hour run while at the rates of 900 and 1200 pounds of coal per hour for the three furnaces together, very steady operation with fairly good intermixture of air with the pulverized coal was secured, and normal stack temperatures prevailed. These temperatures rose rapidly, however, at the beginning of the 1500 pounds per hour total coal rate. In order to maintain smoke at less than ½ degree Ringlemann, it was necessary to increase the secondary air pressure beyond that which had been anticipated. This pressure was not excessive, but the air entering through the air control registers cast the burning pulverized coal backward toward the furnace rear sheets to such a degree

that the temperature of the latter disproportionally rose, and most of the combustion was occurring in the rear third of the furnaces and in the combustion chambers. The same difficulty, to a slightly lesser degree was also experienced at the 1800 pounds per hour total coal rate. At that rate, the layer of diffused coal was somewhat wider and thicker and the intermixture was slightly improved.

Various adjustments of the air control registers and secondary air pressure failed to remedy this condition and it was apparent that a wider and better diffused cone of pulverized coal together with lower velocity secondary air — or secondary air with a different whirl imparted to it — were necessary to decrease high rear sheet and stack temperatures and increase efficiencies.

To preclude possibility that errors in data or judgment had been responsible for the inferior results at the 1500 pounds total coal rate, conditions were duplicated and check runs made immediately on cleaning the boiler after the close of the 72-hour run. The check runs confirmed the previous observations, and the spiral diffusers previously described were installed before further operation was conducted.

With the spiral diffusers installed in the burner throats and the register air openings increased, the intermixture of the air with the pulverized coal was vastly superior to that which had been previously possible, CO_2 and other variables could be maintained within closer limits, and stack temperatures were reduced. The betterment of all these conditions resulted in much greater steadiness of operation and considerably increased efficiencies.

EFFICIENCY OF THE BOILER WATER HEATING SURFACE.

It will be noted from Figure 19 that the efficiency of the boiler water heating surface alone did not vary greatly during the runs of this test; no changes in other efficiencies produced by changes in diffusers or the coal used are therefore in any way traceable to changes in efficiency of the boiler water heating surface. Curve 1 of Figure 19 is the average curve for boiler water heating surface efficiency obtained at the rates of the 72-hour run and the runs with spiral diffusers. Between the rates of approximately 900 pounds per boiler per hour and 1800 pounds per boiler per hour (4500 to 9250 B.T.U. liberated per square foot B.W.H.S. per

FIG. 19.

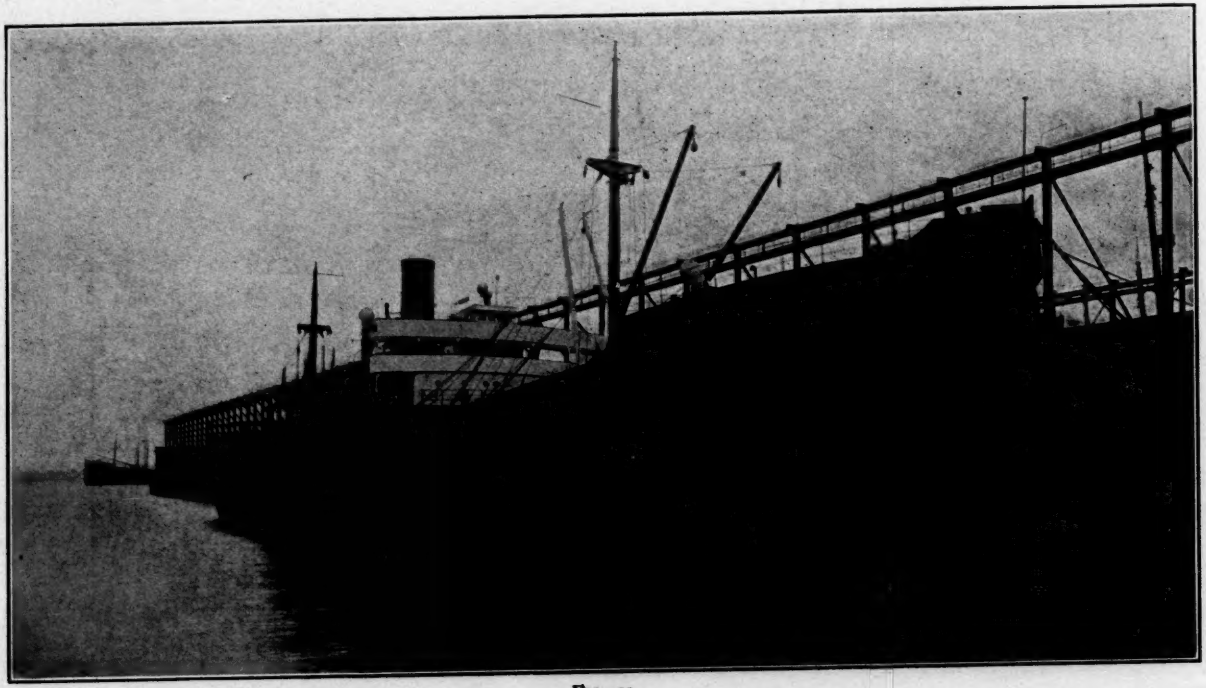


FIG. 20.

hour for Keystone coal), the efficiency of the B.W.H.S. itself alone ranged from slightly less than 99 per cent to just below 96 per cent.

Burning Brazilian coal with spiral diffusers, the actual coal rate was approximately 1080 pounds per hour total or 360 pounds per burner per hour, but the low heat content of the coal made the B.T.U. liberated per square foot slightly less than 3500 per hour. The efficiency of the boiler water heating surface alone at this rate was approximately 99.6 per cent and the spot falls within about 0.2 per cent of the extrapolated curve of B.W.H.S. efficiency. This shows that, in spite of the large amount of ash which accumulated in the lower part of the furnaces during operation with this coal — refer to Figure 16 — the efficiency of the B.W.H.S. taken as an average over the period was not vitally affected, although it was slowly decreasing near the close of the run. At the close of the Brazilian coal run the furnaces and smoke boxes were separately cleaned; and the amount of ash removed was as follows: furnace No. 1, 296 pounds; No. 2, 436 pounds; No. 3, 430 pounds; smoke box No. 1, 206 pounds; No. 2, 301 pounds; No. 3, 320 pounds.

In considering the data shown in Figures 18 and 19 it is to be especially noted that:

The highest furnace and burner efficiency both with the diffusing fins and the spiral diffusers was at the rate of 300 pounds per burner per hour — 1200 pounds per hour total coal rate.

The average combined efficiency of furnaces and burners with the spiral diffusers was more than 5 per cent greater than with the coal diffusing fins and the average overall (boiler, furnaces and burners) efficiency over 4 per cent greater. These figures demonstrate the superiority of the spiral diffusers and increased area for air admission. It is not to be overlooked, however, that the time periods of the runs with the spiral diffusers — shown in Figure 18 — were by no means as long as the duration of runs with diffusing fins. The relatively brief periods of the spiral diffuser runs makes the data taken on them more susceptible to inaccuracies due to lesser number of readings.

The overall boiler efficiency obtained during operation at the 1200 pounds per hour total coal rate with Westphalian coal was

over 5 per cent lower than that obtained with Keystone coal. This was largely due to furnace conditions with the high volatile Westphalian fuel. It is believed, however, that efficiencies with Westphalian coal could have been increased after further experience with it. It will be noted that on Figure 19 the Brazilian coal run spots for overall (boiler, furnaces and burners) efficiency, combined efficiency of the furnace and burners, and combined efficiency of the B.W.H.S., furnace and burners eliminating unavoidable losses, lie considerably above the extrapolations of the curves for the efficiencies obtained when using Keystone coal, while the spot for boiler water heating surface efficiency alone, lies almost exactly on the extrapolated curve for that item. Thus furnace and burner efficiency with this coal were responsible for greater overall efficiency. It is believed that the excellent furnace and burner efficiency obtained is directly traceable to the high percentage of oxygen present in this coal. Thus, in the Scotch boiler furnaces, the high oxygen and volatile contents kept up steady combustion in spite of heavy deposition of ash and inferior grind due to hardness.

The hardness of the Brazilian coal, however, necessitated more power consumption for pulverization, which caused Plant and Fireroom Efficiency spots to fall near the extrapolations of the curves for those efficiencies with Keystone coal.

COMPARATIVE RESULTS.

No comparison of results of this test with those of the Peabody-Kennedy-Van Saun installation (described in the August, 1928, issue of the JOURNAL) is attempted for the following reasons:

The Todd Pulverized Coal Installation was designed for a lower range of rates and maximum efficiency at a lower rate than was the Peabody burner, Kennedy-Van Saun mill, installation. The range of rates run with the subject set up was therefore not identical with that previously covered.

The Todd Pulverized Coal Installation was not run for such long periods as was the Peabody equipment; the conditions of cleanliness of the furnaces and heating surfaces on the two tests were thus not similar.

Neither the burners nor the mills of the two installations can be truly compared with each other, because in the Todd, the burners, mills, primary air fan, and feeders were integral in the so-called "burners," whereas, in the former installations, these functions were entirely separate from each other.

The grind obtained from the Todd pulverizers was much finer than that obtained with the other mill.

The problem of distribution of pulverized fuel was nonexistant with the subject installation, whereas previously it constituted a major problem.

SUMMARY.

Highly promising results were obtained with the installation as originally set up. These were improved, however, by the modifications in design of feeders, coal diffusing devices, and air control registers, which allowed the attainment of steady and uniform feed of coal to the burners and superior intermixture of pulverized coal and air.

Design alterations to render seepage of pulverized coal into mill bearing impossible are being executed by engineers of the Todd Engineering and Drydock Company.

Parts subjected to excessive wear from pulverized coal are being fabricated of metals or alloys capable of much greater resistance to erosion than used on test. This problem has already been partially solved.

The mills as at present constructed require complete disassembly when substitution of pulverizing or wearing members becomes necessary; but it is understood that the manufacturer is making provisions for easier renewal of parts.

The efficiencies obtained with the subject installation compare favorably with those secured with other pulverized coal equipment previously tested.

The relatively high efficiencies during test were due in no small degree to the excellent pulverization obtained. The fineness of grind, produced by the Todd pulverizers over the entire range of rates run, was superior in practically ALL cases to any hitherto secured with other mills when under test at this Plant.

Excellent grind and good results were secured when burning Morrisdale, White Star, Keystone and Westphalian coals with this installation. Slight difficulty was experienced from the coal upon the furnace opening rings when burning the high volatile Westphalian coal — but at no time with this gas coal was the difficulty sufficient to cause unsteadiness of firing or interfere with operation except to a very minor degree.

Good efficiency was secured with the Brazilian (Sao Jeronymo) coal at the rate run. The relatively great hardness of this coal caused the grind to be considerably coarser than that obtained with the four coals mentioned in the preceding paragraph, but the flame conditions were nevertheless good throughout the run. The good combustion obtained with this coal, in spite of its high ash content, was undoubtedly due to its high volatile and high oxygen content. The feasibility of continued operation with this coal, in this type of boiler, depends upon utilization of efficacious ash and soot blowers for cleaning ash from the furnaces. No means were available for thoroughly clearing the furnaces of this boiler during operation, except through the cleanout doors under the burners, which were not used, consequently boiler heating surface efficiency, as well as furnace efficiency, slightly but steadily decreased as the run progressed.

Steady operation with Mindanao coal proved impossible with this installation and boiler as used — the small amount of sample precluded any extensive experimentation which might have made efficient burning possible. The grind of coal obtained was considerably inferior to the Brazilian coal, but this was by no means entirely responsible for the failure to secure steady combustion. The cause of this coal failing to remain alight was principally due to the very low percentage of volatile matter present in it. From the appearance of the flame, it is probable that even this small percentage of volatile burned very slowly, not liberating sufficient heat to maintain the combustion of the relatively high percentage of fixed carbon present in this fuel.

The power consumption of the burners and mills included that necessary for operation of the feeders and primary air fans which are integral with the burners. At the lower rates of 900 and 1200 pounds coal per boiler per hour, the power consumed by this instal-

lation itself (mills, feeders and fans) was but little more than that of previous installations, but at higher rates was considerably greater. The greater power consumption was to a very large degree compensated for by the much better distribution obtained with these burners.

The fact that each burner, mill, feeder, and primary air fan, are all integral with each other in a burner, all burners operating at the same constant speed, with the coal being fed to each burner feeder through chutes in the form of slack, entirely eliminates problems of distribution of equal amounts of pulverized coal to individual burners.

The self-contained nature of these burners makes for flexibility and ease of control; burners may be cut in and out without difficulty. Crippling of one or two burners on the boiler does not necessitate securing the entire installation. Repairs to a burner can be effected while the other burner or burners are in operation.

The installation is inherently well adapted for use with relatively small steam generating units. The boilers of tugs, for instance, could undoubtedly be very satisfactorily fired with these burners.

The weight of this installation and space occupied by it are much less than in the case of other pulverized coal burning installations previously tested at this Plant.

The mills are the source of much less noise during operation than are the ball mills previously tested.

No magnetic separators are used in connection with these burners and mills; tramp iron or other hard fragments are ejected from a safety door in the first stage casing without damage to the working parts.

Coal used, regardless of grade or quality, must be slack, sized, or broken in a suitable crusher prior to passing to the mill feeders. Lumps of even relatively small size cause irregularity of feed and therefore unsteadiness of grinding or firing.

The parts of the burners are interchangeable between burners.

The United States Shipping Board recently awarded the Todd Engineering and Dry Dock Co. the contract for the pulverized coal equipment to be installed on the S. S. *West Alsek*. The installations supplied will be very similar to the one discussed in this article.

the impeller, increased the efficiency to 76 per cent. The change to the diffuser increased the pressure considerably, but there was little increase in volume. While the diffuser is very important in its effect on efficiency, conditions at and in front of the inlet have the preponderant influence on the capacity as well as the efficiency.

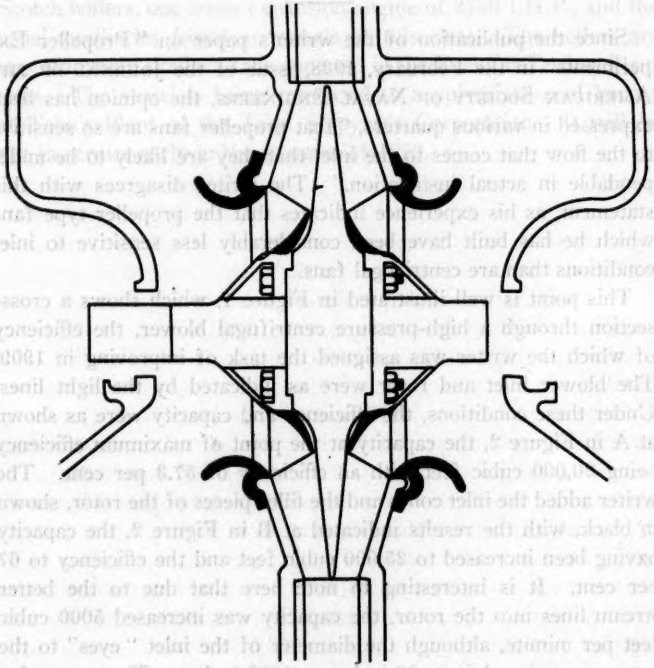


FIGURE 1.

As a further illustration of the sensitiveness of centrifugal blowers to inlet conditions, the cross sections of inlet arrangements of an experimental blower, shown in Figure 4, will be of interest. These figures show a part of the work which the writer did in developing the blowers for the U. S. S. *Barry*. These blowers were described in the February, 1913, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

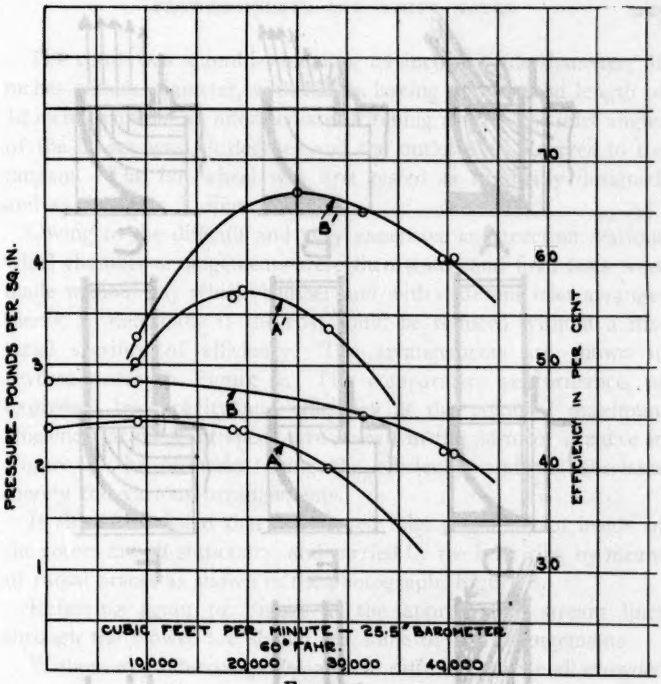


FIGURE 2.

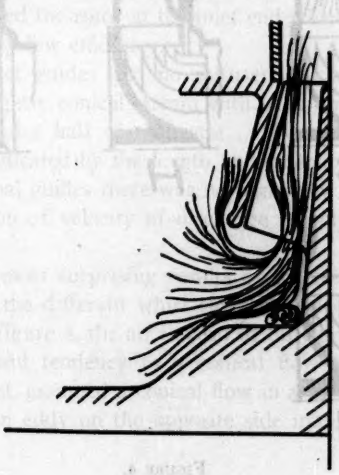


FIGURE 3.

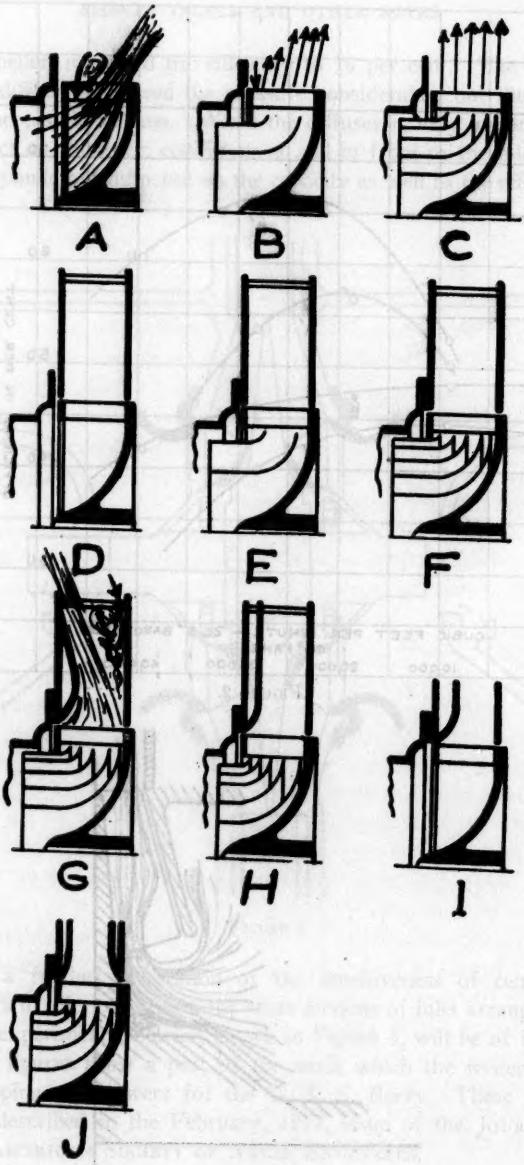


FIGURE 4.

The rotor was a multi-vane fan, 34 inches inside diameter, 39 inches outside diameter, with blades having a free axial length of 10 inches without an intermediate stiffening ring. The inlet angles of the blades was 20 degrees and the outlet 57½ degrees to the tangent. The fan wheel was first tested as originally designed, and as shown at J, Figure 4.

Owing to the difficult and very expensive construction, various whirl chamber arrangements were then tested and final tests were made without any whirl chamber and with different inlet arrangements, to determine if the cost could be reduced without a material sacrifice of efficiency. The arrangements are shown in reverse order in Figure 4. The comparative performance, as expressed by capacity and efficiency at the point of maximum efficiency, at constant speed, are shown in the form of a curve in Figure 5, the ordinates being the efficiency, and the abscissae merely the various arrangements.

It should be noted that the curved inlet cones shown inside of the rotors are all stationary, and carried by the inlet ring by means of radial braces as shown in the photograph, Figure 8.

Referring again to Figure 4, the approximate stream lines through the blower are shown for some of the arrangements.

Without any internal guides and no diffuser, the air all crowded over to one side and was discharged as a conical stream. Air actually re-entered the rotor at the inlet end which partly accounts for the extremely low efficiency.

With two inlet guides and no diffuser, the air was still discharged in a slightly conical stream with back flow through about the first inch and a half of the rotor. The velocity distribution was about as indicated by the length and direction of the arrows. With five internal guides there was no back flow and substantially equal distribution of velocity of discharge in only a very slightly conical form.

Some of the most surprising results of the tests were the flow distributions in the different whirl chambers tried. For instance, in set-up (G) Figure 4, the air instead of diffusing equally or following its natural tendency to a conical flow towards the side opposite the inlet, assumed a conical flow in a thin stream towards the inlet with an eddy on the opposite side into which there was

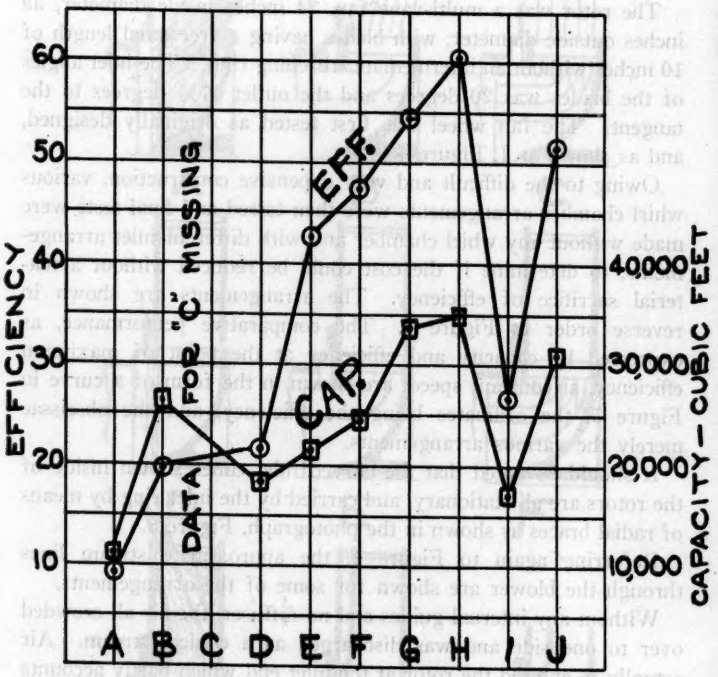


FIGURE 5.

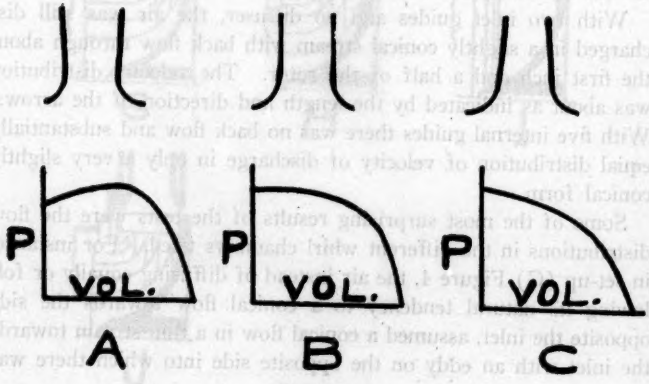


FIGURE 6.

a back flow. This same action was shown with all the "bulge" filler pieces tried in the whirl chamber — the stream always hugging the "bulge" with the greatest curvature.

In general, the effect of vortex chamber arrangements on the pressure volume characteristics of the blower were found to be as indicated diagrammatically in (A) Figure 6. A rapid contraction followed by a divergence of not over 10 degrees total included angle gives a slightly higher maximum efficiency and tends to give somewhat higher pressures at the larger capacities, but the pressure curve has a "hump" in it which is extremely undesirable for blowers which must operate in parallel.

At (B) Figure 6, the passage is parallel beyond the contraction which results in a very gradual sloping characteristic with a quite flat topped efficiency curve only very little lower than the peak of (A). This is generally the most useful and desirable combination. The efficiency, and the flatness of the top of the efficiency curve depend upon the amount of the contraction and spacing of the walls of the whirl chamber.

Gradually contracting the walls as at (C) Figure 6, will produce a more sloping characteristic where this is required for any reason, but the change of slope is purchased at a sacrifice of efficiency and reduction of maximum capacity.

The cross sections of the blowers built for the *Barry* are shown in Figures 7 and 8. It will be observed from the comparative curve of Figure 5, that the efficiency obtained without the inlet cones and with a whirl chamber similar in proportions to those generally employed in blowers at that time was about 26 per cent; whereas, the efficiency obtained on the official Government tests of the *Barry* blowers was 57 per cent, and the steam economy of 1.25 pounds per thousand cubic feet of air delivered with 5 inches static pressure has not been equalled to the present day by any direct connected centrifugal blower of the same dimensions and operating at the same capacity pressure and revolutions with 300 pounds steam pressure, and 10 pounds back pressure.

It will be noted in Figure 7, that the inlet cones are spaced successively farther apart as the radius of curvature of the cones increased. This was done in order to keep as nearly as possible the same ratio of stream thickness to radius of curvature since this

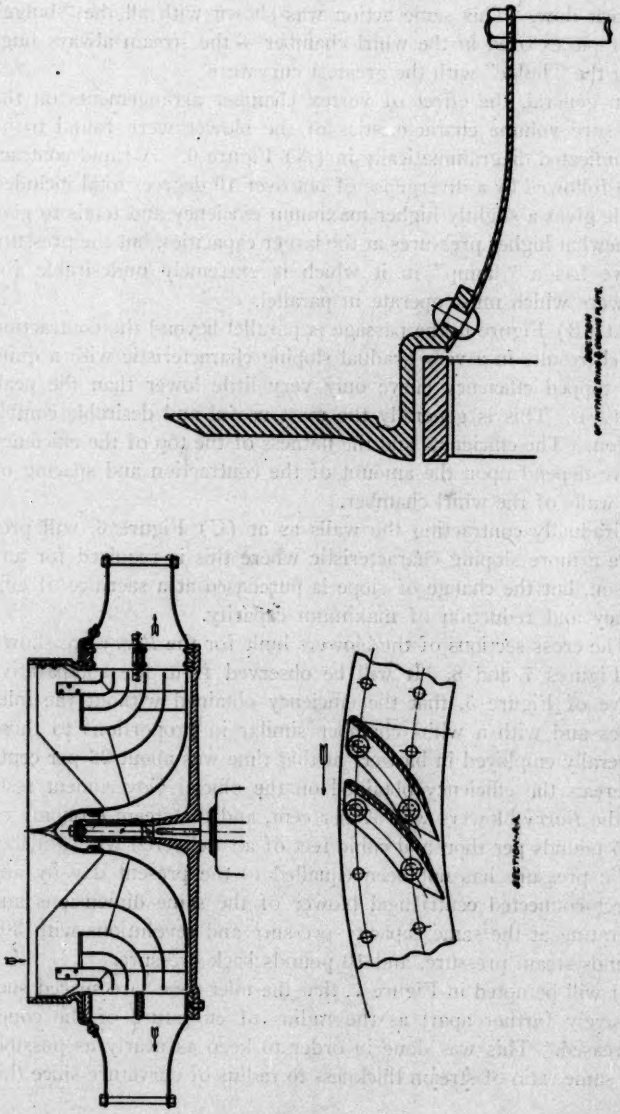


FIGURE 7.

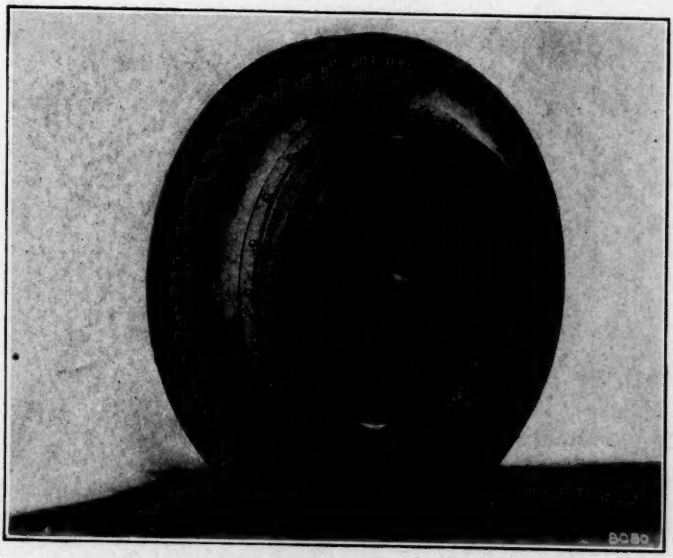


FIGURE 8.

reduces the loss to a minimum. This ratio of T/R is generally kept around .3 where the design permits.

The capacities shown in the tests of Figures 4 and 5, were made by measuring the air at the inlet by means of the writer's standard test nozzle, shown in Figure 9. This form of nozzle gives a flow co-efficient of from .97 for nozzles of 1¼ to 1½ inches to .99 for nozzles of 30 to 40 inches diameter. The design is cheap to build in any size, and is such as to lend itself readily to the measurement of large volumes. The results obtained by this method have been checked at Annapolis againts a 56-inch Venturi meter with substantial agreement between the two methods.

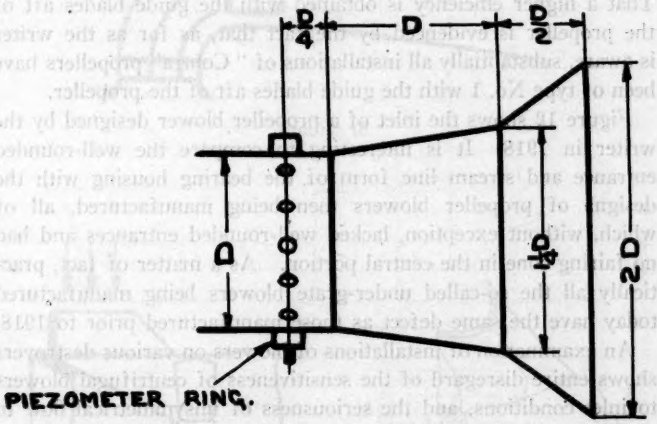


FIGURE 9.

A form of inlet for a propeller blower where radial inlet is necessary is shown in Figure 10. It will be observed that guides similar to the inlet guides shown in Figure 8 are used and the entrance is rounded.

This design may be employed where guide vanes are used on the inlet side, as indicated in Figure 10, the guide vanes in each section between the cones being set at the proper angle to obtain the directional inlet conditions necessary where no guide vanes are employed on the discharge side. Unless these vanes are made mova-

ble, the efficiency curve has a very sharp peak. It will be observed that the curved inlet guides are spaced so as to maintain a substantially equal value of stream thickness to radius of curvature. This inlet may be contrasted with Figure 11, which shows a propeller blower with inlet guide vanes. Note the lack of a rounded inlet and of stream lining to assure parallel stream lines at the entrance to the propeller. The air enters only as indicated by the arrows.

The use of guide blades on the inlet side of a propeller is very old, but came into prominence within recent years, as this form was re-introduced and is known as the "Star Contra Propeller," type No. 2, type No. 1 having the guide blades aft of the propeller. That a higher efficiency is obtained with the guide blades aft of the propeller is evidenced by the fact that, as far as the writer is aware, substantially all installations of "Contra" propellers have been of type No. 1 with the guide blades aft of the propeller.

Figure 12 shows the inlet of a propeller blower designed by the writer in 1918. It is interesting to compare the well-rounded entrance and stream line form of the bearing housing with the designs of propeller blowers then being manufactured, all of which, without exception, lacked well-rounded entrances and had no fairing cone in the central portion. As a matter of fact, practically all the so-called under-grate blowers being manufactured today have the same defect as those manufactured prior to 1918.

An examination of installations of blowers on various destroyers shows entire disregard of the sensitiveness of centrifugal blowers to inlet conditions, and the seriousness of unsymmetrical flow to the rotor. In many instances the intakes have an offset just before the inlet to the blower, apparently with the sole object of maintaining a symmetrical deck arrangement of the inlets without regard to the consequences.

Figure 13 shows a diagrammatic cross-section through the inlet of the blowers located under the galley of many of the destroyers. The intake is in the galley pasageway and there is a sharp offset approximately one-half the diameter of the fan inlet as shown. Assuming that it was impossible to locate the blower in a different position, it would still have been possible to get an entirely satisfactory inlet by employing the arrangement shown in dotted lines

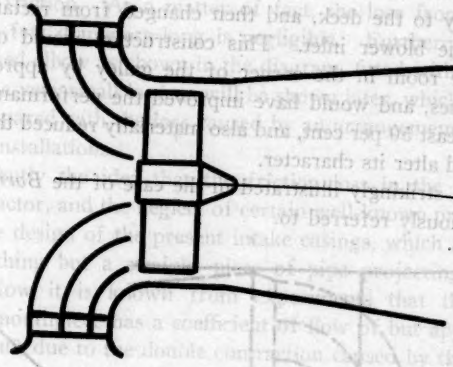


FIGURE 10.

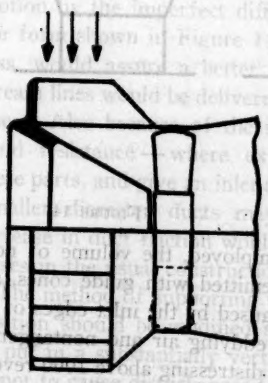


FIGURE 11.

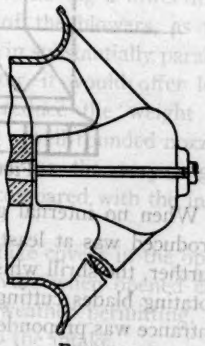


FIGURE 12.

in Figure 13, extending the inlet duct straight up from the fan inlet and opening into the galley passageway with an elbow fitted with "splitters" as shown. The inlet is rectangular and carried rectangularly to the deck, and then changed from rectangular to round at the blower inlet. This construction would only have reduced the room in the corner of the galley by approximately 18×36 inches, and would have improved the performance of the fans by at least 50 per cent, and also materially reduced the volume of noise and alter its character.

This was strikingly illustrated in the case of the *Barry* experiments previously referred to.

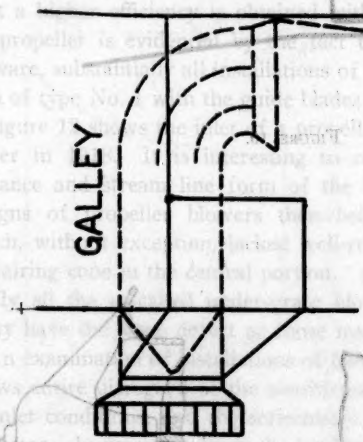


FIGURE 13.

FIGURE 14.

When no internal guides were employed, the volume of noise produced was at least double that emitted with guide cones, and further, the shrill whistling sound caused by the inlet edges of the rotating blades cutting through the eddying air and nontangential entrance was preponderant and very distressing above 1000 revolutions per minute. As built, the *Barry* blowers were indisputably by far the least noisy ever constructed. Incidentally, how the expression "semisilent" ever came to be applied to any of our Naval blowers is difficult to understand unless it was the work of a humorist or was done as an experiment in suggestive psychology.

It will be observed that, as shown by the dotted lines in Figure 13, the inlet duct has only a very slightly larger area than the fan inlet and it may be supposed that this would cause a serious loss through friction. As a matter of fact, the loss from friction in a pipe a few diameters long is negligible. Furthermore, a well-constructed elbow as shown in the diagram, fitted with "splitters," has only a very small loss, as will be shown later, which is insignificant compared with the loss caused by an arrangement such as the present installations.

Apparently the idea that the friction loss in the intakes is a serious factor, and the neglect of certain well-known principles, has led to the design of the present intake casings, which are substantially nothing but a straight piece of pipe projecting above the deck. Now it is known from experiments that the so-called Borda's mouthpiece has a coefficient of flow of but approximately 53 per cent, due to the double contraction caused by the air entering turning an 180-degree angle around the edges of the inlet. Consequently a very disturbed flow to the blower is caused, since the actual entering stream of air occupies only approximately 53 per cent of the area of the inlet and is set in violent eddying motion by the imperfect diffuser action in the pipe. An inlet of the form shown in Figure 14, in addition to having a lower inlet loss, would assure a better performance of the blowers, as the stream lines would be delivered to the inlets in substantially parallel lines. Also because of the smaller diameter, it would offer less wind resistance — where exposed — and reduce the weight of these parts, and give an inlet approximating a well-rounded nozzle. Smaller diameter ducts may be employed, as the very slight increase in duct friction would be trifling, compared with the inlet losses in the usual construction.

The method of supporting the present intake covers in the open position should be modified so that the covers when opened can be put in a substantially vertical position, weather permitting, so as not to cause disturbances to the flow into the intake.

When it is necessary to use elbows or bends in the inlet ducts, these should either be of very large radius of curvature, or if of short radius, should be fitted with "corner blades" as in Figure 16, or "splitters."

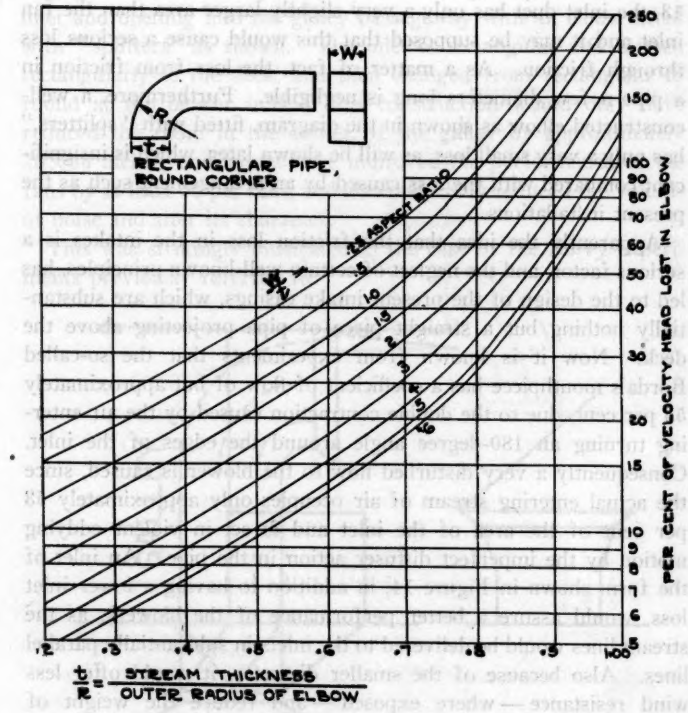


FIGURE 15.

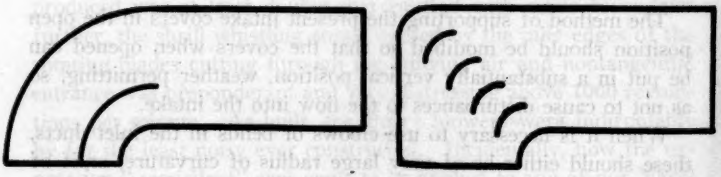


FIGURE 16.

Mr. Loring Wirt in the June, 1927, issue of *General Electric Review* presented what the writer believes to be the most comprehensive set of data on the resistance of elbows which has appeared, and made a very valuable contribution to the art. This article should be of interest to all Naval Engineers because of the extensive use of bends and elbows in the ventilating systems of ships.

The writer has combined Figure 3 and Figure 12 of Mr. Wirt's paper in the form of a curve, Figure 15, which shows the percentage of the velocity head lost in elbows having various ratios of stream thickness to radius of curvature, and different aspect ratios, the aspect ratio being the width of the passage divided by the stream thickness as indicated by the diagram in Figure 15.

By the use of this curve, the loss in any elbow can be found and the loss in elbows fitted with " splitters" can also be calculated. Mr. Wirt, in his paper, gave an example of the reduction of elbow loss by the use of a " splitter" in a square pipe elbow in which the " splitter" was located in the center of the passage. From the curve presented, it will be found that locating the " splitter" in the center of the passage does not give the minimum resistance, which is obtained when the " splitter" is nearer the side with the smallest radius of curvature. For example, if we take a rounded elbow in a 6-inch square pipe, the inner radius being 1 inch, we find that the minimum loss occurs when the " splitter" is 2¼ inches from the inner wall, in which case the loss is 23.9 per cent, whereas with the " splitter" in the middle or 3 inches from the inner wall, the loss as calculated by Mr. Wirt is 29 per cent.

By the use of two " splitters" located 1½ and 3¼ inches respectively from the inner wall, the loss is reduced to 11.9 per cent. In each case, of course, these losses must be multiplied by 1.10 to 1.15 to account for edge losses and additional friction surfaces.

The loss in a bladed corner of any size which is of the form of Figure 16 is given by Mr. Wirt's experiments as approximately 22 per cent. The writer believes that the " splitter" is a more desirable form for Naval vessels, as fewer small pieces are involved and also it is necessary to support the blades of a bladed corner on shrouds, the thickness of which will add additional resistance, as it would be impractical to make all the corners with depressions to bury the shrouds. In general, for ordinary duct work one

"splitter" is sufficient except for very sharp corners, and except where an elbow occurs less than three or four diameters from the inlet of a blower. The general suggestion may be made that where the value of $\frac{t}{R}$ is greater than .75, bladed corners should be used; where $\frac{t}{R}$ is less than .75 and greater than .6, use two "splitters," and for values of $\frac{t}{R}$ less than .6 and greater than .35, use one "splitter"; where the $\frac{t}{R}$ for the whole pipe is less than .35, no "splitter" is required.

If an elbow must be located less than one diameter from the inlet of a blower, a bladed corner should be used, for while its resistance is greater than that of an elbow with two "splitters," it gives a more uniform distribution of flow, which at the inlet of a blower will more than compensate for the slightly greater apparent loss.

Another source of loss in the intake to the blower of Naval vessels is caused by the armor gratings. As usually designed, these have been of rectangular section with sharp corners.

The writer made up a full-size wooden model of the armor gratings for the new cruisers. These gratings are 6 feet square with bars space 3½ inches apart.

At a capacity of 105,000 cubic feet per minute, the loss through the grating was found to be 1 inch, or 12½ per cent of the static pressure required.

As a result of these tests the grating bars are being made with circular tops and bottoms which will reduce the loss by 30 to 50 per cent. If it were practical to roll these bars of stream-line form with circular entrances and a gradually tapering after body, the loss could be reduced to not over 20 per cent of the loss of the original square-corner design.

Of course, it is not always possible to obtain the ideal design of ducts, inlets, etc., but a careful consideration of the few fundamental principles may materially improve performance without inconvenience or a material increase in costs.

As a result of bitter experience there is a general belief among shipbuilders and Naval Engineers that blowers do not give the same performance installed as they do on test, and this difference is quite generally attributed to leakage in the fireroom, boiler casings, air locks, escapes, etc. Pipe friction, armor gratings and reverse flow dampers are also charged with a minor part of the discrepancy. Intake screens do not figure in the difference, as they are supposed to be in place during the acceptance tests.

Running a "soap test" to discover leaks too small to feel with the hand is quite a usual procedure before a full power trial, as though such small leakages could possibly have a measurable influence on the performance of fans having a total credited test capacity of 135,000 cubic feet at 7 inches static pressure at 1500 revolutions per minute. And, in one instance at least, delivered but 50,000 cubic feet at 6.5 static pressure at 1730 revolutions per minute, and on another ship 51,000 cubic feet at 6.75 inches at approximately 1600 revolutions.

The answer to these almost unbelievable discrepancies between the capacity determined by test and the installed performance is not due to leakage in the fireroom, but is accounted for by the sensitiveness of centrifugal blowers to inlet conditions and internal leakage in the blower itself, due to the fact that owing to the method of supporting the blowers in the ship, the "working" of the ship necessitates much larger clearances between the fan wheels and the diffusion plates than those maintained during acceptance tests. Also, in some cases, due to lack of supervision during installation, the diffuser plates are not properly spaced, which should not, however, be possible if they were properly designed so as to assure assembly only in the correct positions.

In the case of propeller blowers designed by the writer, the actual performance installed in the ships indicated slightly better results than were obtained on test at Annapolis, and such losses by leakage as existed (air could be felt coming out around all the hatch covers) were so slight as not to be observable in the results.

It is suggested that the Navy specifications for blowers should contain a clause stating that the performance of blowers as installed must check within not to exceed 2 per cent of the tests as run at Annapolis, with proper penalties for nonperformance.

Also, it is suggested that the Navy specifications should state the minimum clearances between the rotating and stationary parts of the fan based on experience in actual practice, the clearances being sufficient to allow for the working of the ship, normal eccentricity of the rotors, unbalance, etc., this minimum clearance to be used on acceptance tests.

If these clauses were added to the Navy specifications, the present discrepancy between test and practice would disappear. The old leakage excuse could no longer be resorted to, and the fan manufacturer would then be compelled for self-protection to see that the fan inlets and general installation are satisfactory to him, in order that the blower when installed will conform to the specifications.

AVIATION FIRE PREVENTION IN THE NAVY.

By Lieutenant Commander C. G. McCord, U. S. N., Member,
Superintendent of Tests, Naval Aircraft Factory.

As an element of safety in aircraft it is essential that some form of fire extinguisher be carried and that it be so located as to be readily accessible for operation in emergency.

The engine compartment of the plane, lying forward of the fire bulkhead in most conventional types of plane, is the starting place of most fires and therefore the place where fire protection is most needed. Hot exhaust pipes or other hot engine parts, and electrical equipment, ignition or otherwise, probably cause 95 per cent of aircraft fires, the remainder being traceable to hangar accidents or negligence of personnel on the ground or water.

If the engine compartment can be sprayed with an extinguishing substance that will quench the fire as soon as detected, most of the fire hazard has been eliminated. The propeller slip stream and the translation of the plane through the air tend to force a draft for the fire, and more than ordinary means must be employed for extinguishing the blaze resulting, especially where gasoline has become ignited.

Prevention of fires by design of engines and surrounding parts is a proper beginning. Location of carburetors and pipe lines conveying fuel or mixture in relation to hot engine parts and exhaust pipes and to magnetos and electric wiring should receive first consideration, and shielding and bonding of electric conductors should be carried out carefully, so far as weight will permit, to insure grounding without sparking in case of open circuit. Incidentally all magnetos and distributors for use in aircraft should be tested for safety by drenching with gasoline while in operation and should not be installed if they are not so designed as to be safe in this regard.

Fireproofing of fabric covered surfaces, formerly insisted upon in all aircraft, is no longer required. These surfaces are doped to tauten the fabric and in many cases gases are formed in the closed interior of airfoil sections which are highly inflammable and which add to the hazard.

Lightness of the extinguishing equipment, as with all other equipment carried in aircraft, is a prime requisite and the space required for the installation must be minimized. It is also desirable that the equipment be so fitted as to make operation possible from at least two locations in planes carrying more than one person.

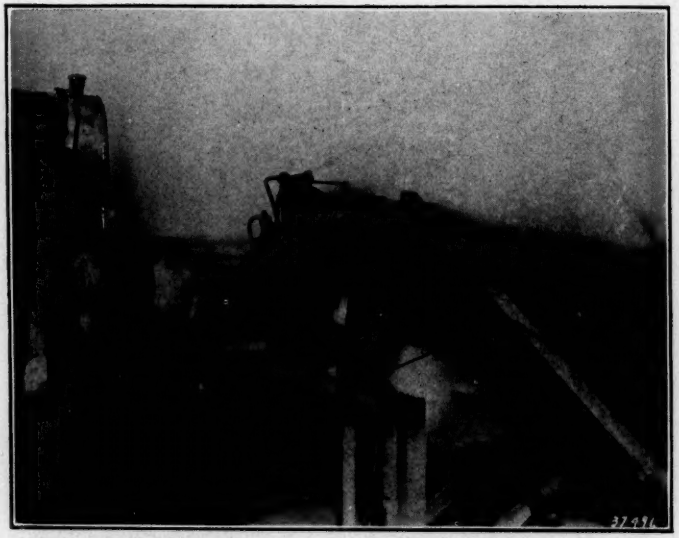
The equipment in itself should be so designed as to require little or no overhaul or attention and constructed of such materials as to insure freedom from corrosion. The extinguishing medium should also be such as not to cause injury to material of the plane or to personnel.

It is desirable that no sparks be left in the neighborhood of the fire that might cause a recurrence of the blaze after it has been once extinguished, as many of the types of extinguisher in use expend all of the extinguishing medium when tripped. A gas, therefore, when used to smother the fire without wetting the surrounding wood or fabric parts of the plane must be either continued long enough or must be cold enough to prevent smouldering or sparks. This of course can not take care of a fire of electric origin where the cause continues after the blaze has been extinguished nor of fires caused by spray of gasoline on hot pipes unless the source of such heat is stopped.

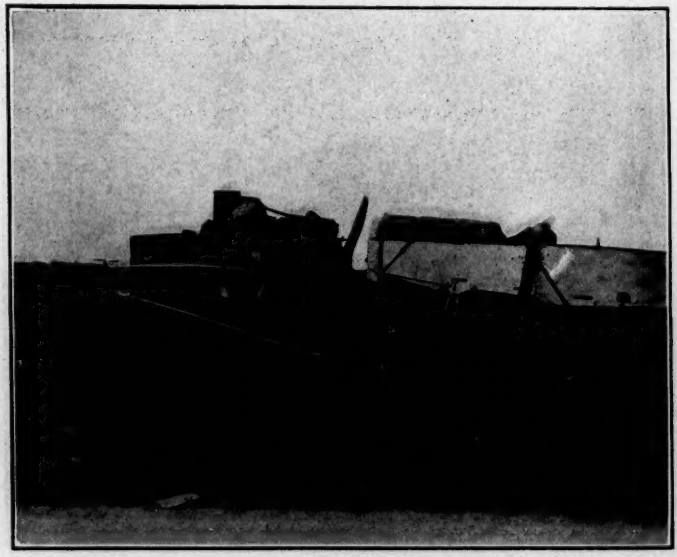
In case of fire in the air most pilots have been instructed to cut the ignition switch and side-slip the plane, thus stopping one source of possible starting of the fire and, in many cases on record, not only keeping the flames from sweeping aft by altering the direction of the air stream, but also actually blowing the fire out. It would seem logical, therefore, to employ this method and in addition an extinguisher system using a gas or liquid under pressure to be sprayed throughout the engine space, particularly around the engine itself, reserving a hand operated liquid extinguisher for use on plane parts not covered by the pressure system and as a further measure in case of recurring fires.

GASOLINE FIRE IN A RADIAL ENGINE USING PROPELLER AIR STEAM. ENGINE NOT COWELED.

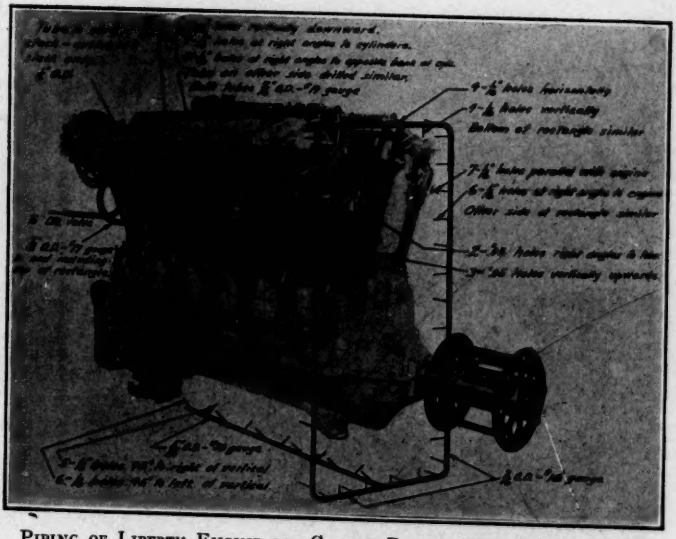
CARBON DIOXIDE SNOW APPLIED TO A FIRE IN A RADIAL ENGINE USING PROPELLER AIR STREAM.



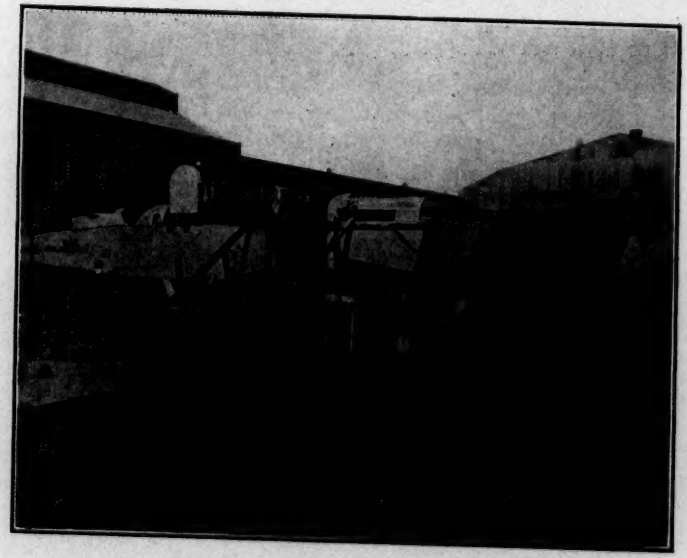
GASOLINE FIRE BEFORE USING EXTINGUISHER. COWELED RADIAL ENGINE.



TEST RIG FOR FIRE EXTINGUISHER TESTS.


9-¼" holes horizontally
7-¼ holes vertically
Bottom of rectangle similar
7-¼" holes parallel with engine
6-¼" holes at right angle to engine
Other side of rectangle similar
2-3/8 holes right angles to line
3-#28 holes vertically upwards
¼ O.D.-#28 gauge

PIPING OF LIBERTY ENGINE FOR CARBON DIOXIDE FIRE EXTINGUISHER.



TEST RIG FOR FIRE EXTINGUISHER TESTS.

As to the materials for use in extinguisher systems, the field at present is somewhat limited. Obviously water or steam are unsuitable on account of the weight involved. Foam and soda acid must be considered from the standpoint of weight and effect on engine and plane parts.

Systems under consideration recently include carbon tetrachloride forced out by air or by carbon dioxide under pressure or by pressure generated by firing a large blank cartridge into a closed chamber and those using carbon dioxide at various pressures and liberated by various means.

The means adopted by the Bureau of Aeronautics of the Navy Department for determining the types of apparatus most suitable for installation on airplanes are, briefly, as follows:

A discarded, "surveyed" airplane fuselage is erected and a discarded engine is mounted in it. Directly in front of this an aircraft engine is set up, equipped with a pusher type propeller so that the propeller blast may be used to simulate the draft encountered by a plane in flight. The old engine is then liberally drenched with gasoline and in some cases a pan of gasoline is placed adjacent to the engine. The propeller is then rotated and the gasoline ignited by spark or torch and the fire is fanned by speeding up the propeller. The apparatus is then operated and the relative efficiency of the extinguishers in extinguishing the fire is noted. Two types of engine are employed with each type of extinguishing apparatus, one a radial and the other a vee type. Reports of tests show the relative weight and efficiency of each brand of apparatus tried.

One aircraft tender carries a dangerous cargo of about 70,000 gallons of gasoline or benzol — the system being so arranged as to provide for carrying both in case it should prove desirable to use benzol or some equivalent, such as ethyl benzene, to replace tetra ethyl lead for the suppression of detonation. This cargo is carried in welded cylindrical tanks in the hold.

The tanks are connected by a system of piping to each other, to expansion tanks which are designed to prevent loss in case of temperature changes, and to outlets at various places where aircraft or boats may be conveniently fueled.

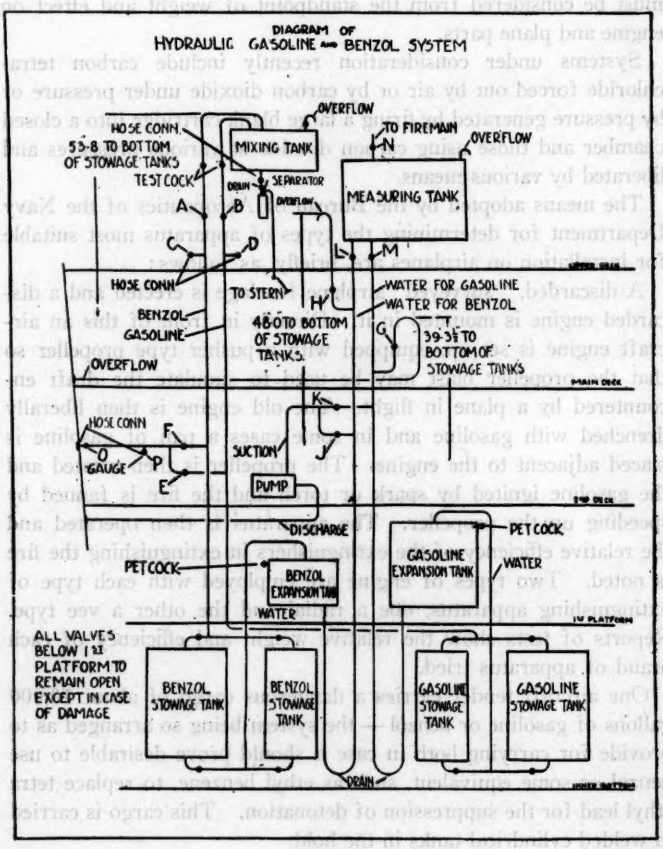


DIAGRAM OF
HYDRAULIC GASOLINE AND BENZOL SYSTEM
OVERFLOW
HOSE CONN.
53-8 TO BOTTOM OF STOWAGE TANKS
MIXING TANK
TO FIREMAIN
N
OVERFLOW
TEST COCK
DRAIN
SEPARATOR
MEASURING TANK
A
B
OVERFLOW
C
D
L
M
HOSE CONN
TO STERN
H
WATER FOR GASOLINE
WATER FOR BENZOL
BENZOL
GASOLINE
46-0 TO BOTTOM OF STOWAGE TANKS
G
39-3½ TO BOTTOM OF STOWAGE TANKS
OVERFLOW
MAIN DECK
K
HOSE CONN
F
O GAUGE
P
SUCTION
J
E
PUMP
DISCHARGE
PET COCK
GASOLINE EXPANSION TANK
WATER
PET COCK
BENZOL EXPANSION TANK
WATER
1ST PLATFORM
ALL VALVES BELOW 1ST PLATFORM TO REMAIN OPEN EXCEPT IN CASE OF DAMAGE
BENZOL STOWAGE TANK
BENZOL STOWAGE TANK
GASOLINE STOWAGE TANK
GASOLINE STOWAGE TANK
DRAIN
INNER BOTTOM

This system is dangerous to the ship and crew from two principal sources: Fumes from leaks in the system, the resulting gas being easily ignited and highly explosive; and fire from the gasoline itself, below, on deck or in planes or boats alongside.

The system is safeguarded from these dangers as follows:

(*a*) Ventilation.— Exhaust and intake blowers and ducts are provided to keep the tank compartment clear of gas fumes from possible leakage.

(*b*) Gasoline being forced out of the system by admitting water from a gravity tank on the weather deck which also serves as a measuring tank and insures that tanks and piping are always full of liquid,— gasoline or water, — and thereby prevents the development and accumulation of explosive gas in the system.

(*c*) A carbon tetra chloride system is installed with tanks located above the storage compartment, to which expulsion air is admitted at sixty pounds pressure, discharging the carbon tetra chloride through perforated ½-inch pipe lines running lengthwise and athwartships of each storage tank. One line is fitted with Grinnell sprinkler heads and the other perforated with pinholes about every six feet.

(*d*) A foam fire extinguisher system is also provided, whereby steam is used to propel foam into the compartment and two twenty-gallon foamite cylinders can also be emptied into the compartment to cover gasoline released by a burst tank or ruptured pipe line.

(*e*) A flooding system for the compartment is likewise installed as in the case of magazines on all combatant ships, for complete flooding with sea water.

(*f*) Hand fire extinguishers and gas masks in the trunk entrance to the gasoline storage compartment complete the safety equipment.

The thoroughness and care exercised in these provisions are an indication of the seriousness with which fire prevention is regarded on board ship.

Where dope is being handled and fabric installed or removed, both pyrene and hand foam extinguishers are kept within reach at all times.

The attached table is self explanatory and is a fair representation of the character of extinguishing apparatus submitted for test.

APPARATUS		A	B	C	D
EXTINGUISHING MED.		CO_2	CO_2	CCl_4	CCl_4
CONTAINER		20 Lbs. capacity Steel bottle with valve	7½ lbs capacity Steel bottle · sheet brass cap.	One quart capacity Air flask. same size	One quart capacity
ACCESSORIES		3' Flex. hose. Manifold. Diffusing Cone	Special Manifold	Special Manifold	Special Manifold
RELEASE		¼ Turn Valve handle.	7/16" diam. hole cut in brass disc	¼ Turn valve handle admits air and releases	210° Turn of handle punctures CO_2 cartridge
WT. COMPLETE		52.5 lbs.	42.5 lbs.	17.5 lbs	17.5 lbs
WT. CONTAINER		With valve 25.5 lbs	22 lbs.	7.5 lbs.	7.5 lbs
TRIALS	VEE	28	69	0	0
	RAD.	50	28	24	27
% SUCCESS	VEE	42	97	–	–
	RAD.	90	96	0	0
LBS. TO EXTINGUISH	VEE	7	7	–	–
	RAD.	3	3	–	–
REMARKS		CO_2 snow and gas are emitted Special manifold installed on Vee engine with 100 #50 drill holes	Bottle emptied in one second	Unsatisfactory	Unsatisfactory

Note - Liberty vee and Kinney-Noble radial engines used in surveyed D H fuselage 80mph slip stream Carburetor floats removed and fuel lines sawed at carburetor connection and 18" distant to emulate broken lines

The tests conducted by the Navy and War Departments are of considerable interest to commercial concerns manufacturing aircraft and to insurance corporations who are entering the growing field of insurance against the various risks encountered in aviation, as evidenced by the volume of correspondence wherein the concerns mentioned have been eager to secure information regarding the results obtained.

Some danger of fire starting from static spark exists when gasoline is strained through a chamois skin strainer into the fuel tank of a plane, the spark being induced by friction. This danger is eliminated by grounding of metal tanks and bonding and grounding of metal pipe lines or hose. This procedure should be followed for gasoline transfer regardless of the method of straining used.

PREVENTION OF AIR-FREEZING IN COLD WEATHER DIVING.

By Commander W. C. Owen, U. S. Navy, and Lieutenant O. E. Grimm, U. S. Navy, Members.

Soon after the start of diving operations to recover the U. S. S. *S-4* during the winter of 1927-1928, trouble was experienced on the salvage vessel, the U. S. S. *Falcon,* in furnishing air to the divers without having the moisture in the air freeze in the air-inlet valve to the diving helmet, and the necessity arose of preventing this freezing in order that salvage operations might continue through the winter.

The *S-4* was sunk on December 17, 1928, in 103 feet of water off Provincetown, Massachusetts, and the weather was the usual winter weather for this locality, with snow, rain, fog, and temperatures as low as 8 degrees F. and with sea-water temperatures about 33 degrees F.

The trouble experienced was a freezing of the moisture condensate as the air was cooled in the air hose to the diver and in the air-inlet valve in the diver's helmet. It was immediately apparent that if the aqueous vapor of the air could be removed before the air was sent to the diver that freezing could not take place. It was also apparent that it was not necessary to remove all aqueous vapor if the dewpoint of the air to the diver was lower than any temperature to which the air could be cooled in any part of the system. Experiments have shown that there is a temperature drop of about 6 degrees F. in the air in passing through the present type of air-inlet valve in the diver's helmet. With water at the diving depth at a temperature of 32 degrees F., and a further drop of 6 degrees F. in the valve, any equipment or method proposed needed to be capable of furnishing air with a dewpoint of 25 degrees F. or lower.

There were several methods available for removing the vapor content of the air, and it was decided that the most satisfactory method was to condense out as much moisture as possible at high pressure, then to expand the air to further lower the dewpoint, and then to heat the air to insure it reaching the diver sufficiently above the dewpoint and freezing temperature to avoid any danger of condensation or freezing. Heated air would also be more comfortable for the diver, although a compromise was necessary between comfort and relative humidity for the diver, as it was apparent that with a fixed dewpoint for air, the relative humidity varies inversely as the temperature.

The *Falcon* was and is equipped with two air compressors capable of a discharge pressure of 150 pounds per square inch. At the start of the diving operations a pressure of 144 pounds per square inch was used — that is, 44 pounds for a depth of 103 feet, plus 100 pounds additional. This was later reduced to 50 pounds additional, so that an air pressure of 95 to 100 pounds was supplied to the diver's air line. The compressors were, however, operated to supply air at 150 pounds pressure, and the air was then expanded as required for the diver. After leaving the compressors the air was passed through a cooler in the firerooms and then sent to the air reservoir, but this was not sufficient to condense out the moisture necessary to give a dewpoint below the temperature at diving depth.

An additional cooler was then installed between the fireroom cooler and the reservoir which was designed to take advantage of two cooling effects — that of the salt water passing through the tubes with the diver's air surrounding the tubes, and the further cooling effect of the atmosphere surrounding and in contact with the outside shell of the cooler. The air entering the cooler was at an average temperature of 90 degrees F. (saturated) and containing 14.79 grains of vapor per cubic foot, and it left the cooler at an average temperature of 35 degrees F. (saturated), containing 2.37 grains of vapor per cubic foot. In passing through this cooler 12.42 grains of vapor per cubic foot were condensed, and this was drained off through a drain from the lower part of the air side of the cooler. Much of the cooling was done through the outside shell of the cooler, which was exposed to temperatures as low

as 8 degrees F., and the temperature of air leaving the cooler was nearly the same as that of the entering cooling water. The condensed moisture, falling on the lower tube sheet, was prevented from freezing by being in contact with a surface exposed to salt water. With this design no trouble was experienced in freezing of condensate except in the drain pipe which could be cleared with a spike. A separator was installed between the cooler and the reservoir to remove any water that might be carried over from the cooler.

After leaving the air reservoir at about 32 degrees F. (saturated) the air was expanded through a reducing valve to the pressure required for the diver, in this case 100 pounds per square inch. Air at a temperature of 32 degrees F. (saturated) contains 2.113 grains of vapor per cubic foot. After expansion at a constant temperature from a pressure of 150 pounds to 100 pounds it contained 1.409 grains per cubic foot, and therefore then had a dewpoint of 22.8 degrees F., and a relative humidity of 66-2/3 per cent. While air with a temperature of 32 degrees F., a dewpoint of 22.8 degrees F., and 66-2/3 per cent relative humidity was found to be safe and reliable for diving purposes, it was believed to be desirable to install an air heater which raised the air to such a temperature that it reached the diver at about 40 degrees F., with a dewpoint of 22.8 degrees F., and with 49.4 per cent relative humidity. This air was found to be very agreeable from the standpoint of comfort for the diver.

With the foregoing equipment and method of operation, the *Falcon* was able to continue salvage work until the *S-4* was raised in March, 1928, and towed to the Navy Yard, Boston, Massachusetts.

In general, the following factors enter in all problems of air conditioning or dehydration for diving purposes:

(*a*) *Depth of water.*—Greater air pressures are required for increasing depths. In general this will be .45 times the depth of water in feet, plus 50 pounds additional.

(*b*) *Temperature of water at diving depth.*—If this temperature is below 40 degrees F. there will always be danger of freezing, unless the dewpoint is at least 10 degrees F. below the temperature of sea water at diving depth.

(*c*) *Air pressures required for diver.*—This is determined as in (*a*) above.

(*d*) *Dewpoint of air to diver.*—This should be at least 10 degrees F. lower than the temperature of water at diving depth.

(*e*) *Amount of air required.*—This is dependent on the number of divers operating at the same time, together with other needs for air.

(*f*) *Temperature of sea water.*—Colder injection gives greater cooling effect in all coolers, with consequent lower dewpoint of air to divers. On the other hand, colder sea water increases the danger of freezing.

(*g*) *Atmospheric air temperatures.*—Colder atmospheric temperatures give better cooling effects in the main coolers. Cooler air to the compressors also requires less cooling after leaving the compressors.

(*h*) *Relative humidity of atmosphere.*—Lower humidity means less dehydration required. As the purpose of any such installation is dehydration, it follows that dry air can be more easily conditioned than air with a higher humidity.

(*i*) *Humidity and temperature of air to diver.*—It is apparent that with a dewpoint low enough to prevent condensation and freezing that there must be a compromise between these two factors. Any heating with a fixed dewpoint lowers the relative humidity.

In analyzing problems of air conditioning for diving, or in designing equipment for purposes of dehydration, similar to the installation on the *Falcon,* the foregoing factors would be considered as follows. Any installation of this kind must be capable of performing its allotted function under the most extreme conditions probable.

Such extreme conditions might be stated to be—

Atmospheric temperatures as low as 0 degree F.

Sea water temperatures as low as 27.5 degrees F.

Diving depths of 300 feet.

For a depth of 300 feet a diver would be given an air pressure of $300 \times .45 + 50 = 185$ pounds per square inch. With sea water temperature at 27.5 degrees F., and allowing a margin of 10 degrees F., would require a dewpoint temperature of 17.5 degrees F.

From data obtained on the *Falcon* it is possible to predict a temperature of air from the last cooler as low as 35 degrees F. Therefore the air to the diver must be at a pressure of 185 pounds, must have a dewpoint of 17.5 degrees F., and will therefore contain not more than 1.103 grains of vapor per cubic foot. The air from the last cooler before expansion to the diver's air line will be at 35 degrees F. (saturated), containing 2.366 grains of vapor per cubic foot. Therefore the expansion ratio must be $\frac{2.366}{1.103} = 2.15$, and the compressors must deliver air to the reducing valve at $2.15 \times 185 = 398$ pounds per square inch. The coolers must be designed to carry a working pressure of 400 pounds, with sufficient cooling surface (air and water) to produce a cooling effect as low as 35 degrees F. The air heater must be capable of heating the air to temperatures as high as 200 degrees F. Experience on the *Falcon* showed that the cooling effect in approximately 600 feet of diver's hose immersed in sea water at 33 degrees F. necessitated a temperature of 180 degrees F. for air leaving the heater to reach the diver at 40 degrees F. Drains and separators must be capable of handling the condensate, and relief valves must be installed as a safety feature. Air to the diver at 40 degrees F. with a dewpoint of 17.5 degrees F. will have a relative humidity of 38.7 per cent, which should cause no discomfort nor ill effects to the diver.

The installation on the U. S. S. *Mallard*, designed by the Boston Navy Yard, and recently completed at that yard, is designed to cover extreme conditions of temperature and depth of water. The equipment consists essentially of the following:

2—Air compressors — each capable of delivering 170 cubic feet of free air per minute at a pressure of 400 pounds per square inch, located in the after fireroom.

2—Accumulators or air receivers — each of 18.8 cubic feet capacity, installed in the after fireroom to absorb the air pulsations from the compressors.

2—Air coolers, using salt water for cooling medium, each of 50 square feet cooling surface, located in the after fireroom.

1—Main air cooler, using salt water and atmospheric air as cooling medium, of 80 square feet cooling surface, located on the boat deck.

1—Main air receiver or reservoir, of 110.5 cubic feet capacity, located on the boat deck.

1—Reducing valve capable of adjustment between 200 and 75 pounds, with an initial pressure of 400 pounds per square inch.

1—Air heater, located in the fireroom hatch way, under the boat deck.

1—Torpedo air compressor, capable of delivering air at 3000 pounds' pressure to a high pressure air bank, consisting of 30 units of approximately 5 cubic feet capacity each. The torpedo air compressor is located in the engine room and the air bank in the forward fireroom.

1—Recompression chamber, located on the starboard side of the main deck.

2—8½-inch cross-compound air compressors, capable of delivering 150 cubic feet of free air at 140 pounds pressure for salvage purposes — located in the forward fireroom.

Also separators, drains, relief valves, and gauges and thermometers for recording data.

The two coolers in the fireroom are designed to pass the air through the tubes with the cooling medium surrounding the tubes, while the main cooler on the boat deck is similar to that on the *Falcon;* using water through the tubes with diver's air surrounding the tubes, and atmospheric air on the outside of shell. No cooling water is provided in the main air receiver, as was the case on the *Falcon.* This installation has not yet been put to a severe service test, although the Navy Yard tested the installation before its departure from Boston in December, 1928. The results obtained, while not conclusive in themselves, are sufficiently illuminating in the light of past experience on the *Falcon* to give promise of excellent results under the normal operating conditions.

STEAM TURBINE ELECTRIC DRIVE FOR MERCHANT VESSELS.

By H. C. Berrian, Assistant Chief Engineer Newport News Shipbuilding and Dry Dock Company.

The turbo electric system of propulsion was developed in connection with the vessels of the U. S. Navy. There is no need to go into that at this time as the subject has been fully covered. It has only been within the last two or three years that the system has been applied on a considerable scale to merchant vessels. The first installation of consequence was the Panama Pacific vessel *California,* which was built by the Newport News Shipbuilding and Dry Dock Company, and started her maiden voyage January 28, 1928. This vessel was followed by the *Virginia* for the same line, which went into service a year later, and will be followed next fall by a third sister vessel, the *Pennsylvania.* In the meantime, the system has been applied by the British to the *Viceroy of India,* a vessel of generally similar characteristics and power to the Panama Pacific vessels above noted.

At the present time several other vessels with a similar type of propelling machinery are under construction in this country, including those for the Grace Line, the Ward Line, and other vessels which are projected.

It is of interest to inquire into the characteristics of this mode of propulsion. In the first place, it is generally conceded that the first cost of turbine electric machinery normally exceeds the cost of geared turbine machinery for similar duty, *i.e.,* power, steam pressure and temperature, vacuum and propeller revolutions. It is the feeling of many marine engineers that the various types of propulsion machinery still have their proper field. The reciprocating engine with moderate steam pressure is still favored for small and moderate powers when fuel consumption is not of first moment and where simplicity and reliability are demanded by the

class of operators that are available. The geared turbine with its higher steam pressure and temperature now normally claims the field which is beyond the capacity of the merchant type of reciprocating engine or where fuel economy becomes important. As the power of the geared steam turbine increases it gradually becomes increasingly difficult to provide adequate and reliable reduction gears. Undoubtedly, there is still a field in which either geared turbine or turbo electric drive would be satisfactory. Just where this point is, it would be difficult for engineers to agree, but in the opinion of many American engineers there is a point when the essential simplicity of the electrical speed reduction provided by the alternating current generator and motor indicates it as being the most reliable method of obtaining that reduction in speed of rotation between turbine and propeller which is necessary to obtain satisfactory efficiency of each. The principles involved in electrical transmission on shipboard are identical with those which have been thoroughly worked out in central station practice and it is quite natural that the United States, which has been the leader in electrical development on land, should be the leader in electrical development on shipboard.

When we come to turbo electric propulsion, the designer at once finds himself with a free hand in determining the steam conditions that may be used. The single, relatively high speed turbine rotor, constantly revolving in one direction, free from the complications of an astern element with its varying expansions and direction of thrust, has been thoroughly worked out in land practice for pressures and temperatures in excess of anything he will probably care to adopt. The condensers, boilers and piping are the factors which will determine the pressure and temperature to be employed. The adoption of electric propulsion indicates the use of electricity for most of the auxiliaries, and, as an appreciable amount of current must be supplied for excitation, the size of the auxiliary generators becomes such that condensing rather than non-condensing operation is indicated for them with a correspondingly low water rate. In fact, the water rates that are now obtainable with auxiliary turbo generators are so low that it is doubtful whether there is any advantage as far as fuel cost is concerned in installing Diesel drive generators. Steam for feed heating and similar purposes

is obtained simply by bleeding from a suitable stage of the main or auxiliary turbines which does away with the trouble often encountered with erosion of turbine blades when the flow is in the opposite direction, *i.e.*, when non-condensing auxiliary generators and steam driven auxiliaries are employed and the excess exhaust not used in the feed heater is returned to a low stage of the main turbine. Bleeding also relieves the low pressure stages at the expense of the high pressure stages, which tends to improve the turbine efficiency.

The following main features must be considered in addition to those already mentioned, when deciding the type of propelling machinery:

SPACE.

The relative engine room length depends partially on the beam and draft of the ship. The engine room length on the *Virginia* is about six feet less than would have been required for a geared turbine installation. On other types of vessels this relation may be different. Electric propelling machinery may be arranged to give more head room and greater accessibility. The turbo generators with the switchboard are usually located one deck level above the inner bottom. This of itself permits the condenser to be placed below the turbines, making the condenser and its accessories more accessible. Further advantages of this arrangement will be pointed out later.

WEIGHT.

Turbo electric is usually heavier than geared drive until extremely high powers are reached. However, this is due partially to two factors which could be eliminated if desired. The geared turbines are usually designed expressly for the job, whereas the electric propelling machinery, due to the production requirements of the manufacturers, frequently covers a wide range of power, the variation between different installations being obtained by comparatively minor changes in the internal construction of the turbines and in the windings of the generators and motors. The auxiliary generating plant is usually made larger than is strictly necessary for the requirements of the main machinery, as the tendency is to electrify

all auxiliaries possible, even down to the galley equipment and stateroom heaters. Electrification of auxiliaries is highly advantageous in itself due to better arrangement, greater flexibility and better economy; but the increased auxiliary generator weight should not be charged primarily to electric drive.

FUEL ECONOMY.

This is a very prolific subject for argument and often "the wish is father to the thought" when forming an opinion. An impartial study of the operating records would seem to show that there is no appreciable difference between fuel consumption of geared turbine and turbo electric propelling machinery of the same power, *provided the results are corrected to the same steam conditions, the same type of draft, the same combustion air and feed water temperatures, the same vacuum and the same propeller revolutions, and if the same safety factors for blade stress and vibration are maintained.* At present we have no accurate *comparative* data of similar installations. Most of the service results are in fuel per S.H.P. per hour and a slight variation in boiler efficiency will wipe out any difference in turbine efficiency. Suffice it to say that the owners of either type of drive are usually satisfied with the operating results of the decision which they made when selecting the drive at the time the ship was being designed.

It should be noted that owing to the gear teeth and the multiplicity of high speed bearings the geared drive requires from three to four times as much lubricating oil, thus necessitating larger pumps, coolers, tanks and piping, as well as a higher operating cost for lubricating oil.

RELIABILITY.

This question has now been settled in so far as electric drive is concerned. With the experience gained in previous installations the manufacturers are now making their marine propelling machinery just as reliable as their central station units, and the latter need no defense.

Electric drive enjoys one advantage over geared drive from the standpoint of reliability. If one main generating unit is disabled

the other main generator can supply current to both propelling motors, giving a speed corresponding to something less than half power or about 70 per cent. It is true that substantially the same result can be obtained with geared drive provided only one of the turbines or pinions is disabled. However, this necessitates the installation of temporary cross over pipes (carried for the purpose) or changing from blank to ring flanges, to effect the emergency connections. This changeover requires considerable time. With electric drive the change to one generator is simply a matter of switching, involving no delay.

OPERATION.

Engineers capable of operating a geared job have no difficulty in handling the electric installation, although a higher class chief electrician is usually carried. Very little, if any, greater technical knowledge is required.

Furthermore, the control board meters furnish power data at all times. In fact, the engineer on watch knows the power the machinery is developing without making any conscious effort to get it. This knowledge is automatically obtained with ordinary routine operating. How different the occasional trip to the shaft alley to read the torsionmeters on a geared drive installation! If the torsionmeter gets out of adjustment it is very liable to stay that way as S.H.P. readings are not vital to operation. Knowing the power, the fuel per S.H.P. can be readily calculated from the oil tank soundings. Faulty operation or defects in machinery, such as excessive shaft bearing friction, damaged propeller blades, poor steering, are readily detected from the power meter readings.

COMFORT OF PASSENGERS AND CREW.

From this standpoint electric drive has no equal. Vibration is reduced to a point where the critic must search for it. Noise is also reduced to a minimum. The noise of the generator ventilating vanes is low pitched and is not nearly so objectionable as the high pitched tone of some reduction gears. Racing of the propellers is eliminated due to the turbine governor holding down the speed of generator, turbine and motor.

AUXILIARIES.

As has been mentioned above, on an electrically propelled vessel the natural tendency is to drive the auxiliaries electrically wherever possible. This is desirable from almost every operating standpoint, but particularly from that of economy. With motor drive each auxiliary, whether large or small, is supplied with current generated by turbo generators having a water rate of about 20 pounds or less per K.W. Assuming an average motor efficiency of 85 per cent the rate per S.H.P. will be about 32 pounds as against 80 to 120 pounds for the reciprocating type. Of course, with reciprocating auxiliaries the exhaust would be used to heat the feed water whereas the auxiliary turbo-generators operate condensing at sea as well as in port. However, feed heating is accomplished by bleeding as noted above. Leakage of steam to idle auxiliaries is also eliminated, which further improves the overall economy. It is often considered advisable to use turbine driven main feed pumps and steam reciprocating fuel oil pumps on account of the simplicity of speed control; and steam reciprocating lubricating oil pumps are generally preferred because of their extreme reliability, which is necessary for the oil operated turbine generators.

The motors driving the auxiliaries are ordinarily direct current, but for the high powered liners of the near future the tendency will be towards the use of alternating current motors wherever a wide range of speed control is not required, as they are of simpler design and require less attention. Propulsion motor ventilation fans, forced draft blowers, main circulating and other pumps, steering gear, galley, lighting, passenger and cargo ventilation fans would use alternating current. The windlass, cargo winches and boat hoists would ordinarily use direct current supplied by the excitation generators which are idle in port. However, for certain types of ships, alternating current should be used for these also.

DUAL DRIVE.

Another method of improving the economy of the auxiliaries is by means of the "dual drive" system. This is used where it is desirable to obtain the highest degree of economy. The auxiliary generating sets are of the three-unit type consisting of a turbine,

a direct current generator and a synchronous motor. When the vessel is operating above two-thirds speed current from the main generators is supplied to the motor which drives the direct current generator. The auxiliaries receive their current from this generator, the voltage of which is kept constant by a voltage regulator, thus obtaining practically the same water rate as the main propelling units. When the main generator speed drops below two-thirds of its normal value, as in maneuvering, the auxiliary turbine automatically picks up the load and drives the auxiliary generator. Numerous variations of this system may be used. When some of the auxiliaries use alternating current they may be supplied direct from the main generators through a transformer. Auxiliary turbine drive must, of course, be provided for speed conditions below two-thirds of normal.

CONDENSING PLANT.

As shown by central station practice the proper place for the condenser is directly underneath the turbine exhaust opening. There are two important reasons for this. First, it furnishes perfect drainage for the turbine without the use of any accessories, such as pumps or injectors. Second, the pressure drop in the long, curved, internally stayed exhaust pipe is eliminated. This means, for the same conditions, an increase in vacuum at the turbine of from .25 inch to .5 inch of mercury or a reduction in water rate of from 1½ to 3 per cent. This is particularly valuable where the vessel is operating in hot water.

With a properly drained turbine, high vacuum ejectors and condensate pumps can be used, thereby saving weight, space, cost and steam, and also allowing duplication of apparatus which is not feasible with steam reciprocating air pumps and vacuum augmenters.

A further advantage of air ejectors over air pumps and augmenters is the elimination of air from the feed water. The air extracted from the condensers by the ejectors is vented to the atmosphere, whereas the air removed by the augmenter is carried to the air pump where it is further compressed and an emulsion of air, non-condensable vapors and water is discharged to the feed tank. From the feed tank it goes to the boilers and unless special

deaerating apparatus is installed, corrosion of the boiler tubes is very likely to result. With high temperature, high pressure installations the use of ejectors becomes practically a necessity. This is indicated by their universal adoption in up-to-date central station installations.

S.S. "VIRGINIA."

Captain Roger Williams, Operating Manager of the International Mercantile Marine Company, in his paper "Description and Trials of the S.S. *California,* delivered at the November, 1928, meeting of the Society of Naval Architects and Marine Engineers, has given an admirable description of the vessel. It will be interesting to note some of the improvements adopted on her sister ships, the *Virginia,* now in service, and the *Pennsylvania,* still on the ways. The plates accompanying this paper show the engine room arrangement on these two Panama Pacific liners. Particular attention is called to the accessibility of main units and auxiliaries.

Like the *California,* the main propelling machinery of her two sister ships consists of two turbo generators which furnish current for two propelling motors, each driving one propeller. Three-phase alternating current with a maximum voltage of 4000 is used. The propelling motors are of the General Electric Company's synchronous induction type, each having a maximum continuous rating of 8500 S.H.P. at 120 R.P.M. The generators, each having a maximum continuous rating of 6600 K.W., are direct connected to marine type turbines having a maximum capacity of 8500 propeller S.H.P. each at 2880 R.P.M. Normally each generator furnishes current to the corresponding motor, but the switching mechanism is so arranged that one generator can supply current to both motors, delivering somewhat less than its rated output, giving the vessel a speed of about 13.5 knots. The generators are ventilated by attached internal fans which draw the air from the generator and force it through a cooler and back to the generator, making a closed circuit. The air coolers are supplied with water from the main circulating pumps. Each propelling motor is ventilated by two independent motor driven fans, which draw the air from the engine room through the motor and discharge it up the ventilating stack.

The main condensing plants are identical with those of the *California*, each consisting of a cylindrical surface condenser having 11,000 square feet of cooling surface and served by twin two-stage air ejectors with after condenser and two 8000-G.P.M. circulating pumps driven by 100 H.P. direct current variable speed motors. There are three motor driven main condensate pumps, one for each condenser and one connected as a spare for either plant.

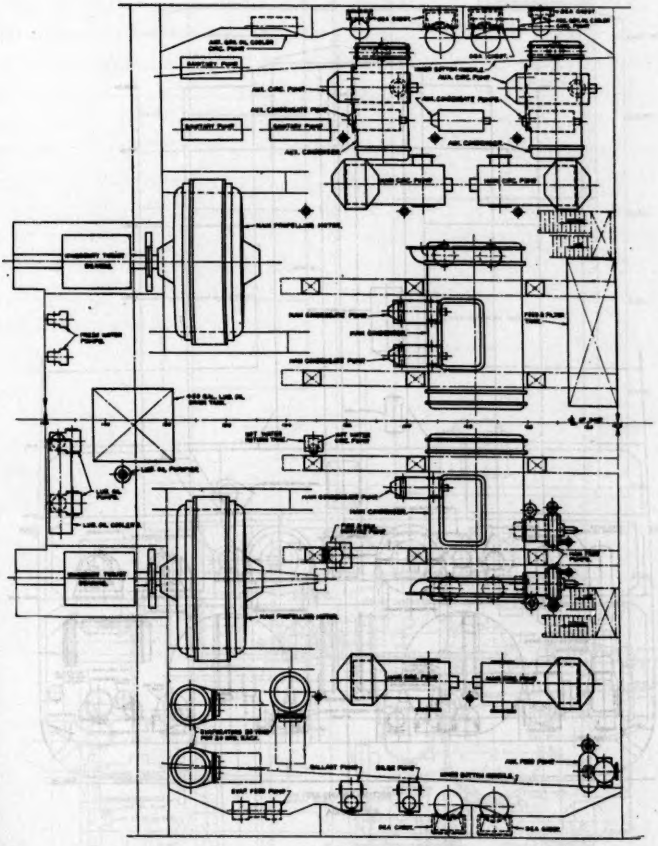
Two-stage feed heating is used as on the *California*. The first-stage heaters take steam at about atmospheric pressure from the main turbines, heating the feed to about 200 degrees F., and the second-stage heater utilizes the auxiliary exhaust, raising the temperature to about 230 degrees F.

Current for excitation, propulsion auxiliaries and ship's lighting and power is supplied by four G. E. 500 K.W. 120/240-volt limited three-wire direct current geared turbine driven generators operating on full boiler pressure and superheat and exhausting into two condensing plants which are small editions of the main condensing plants. Twin air ejectors and a 30-H.P. motor driven centrifugal pump serve each of the two 2800 square feet surface condensers and there are three condensate pumps arranged so that the third pump serves as a standby for either condenser.

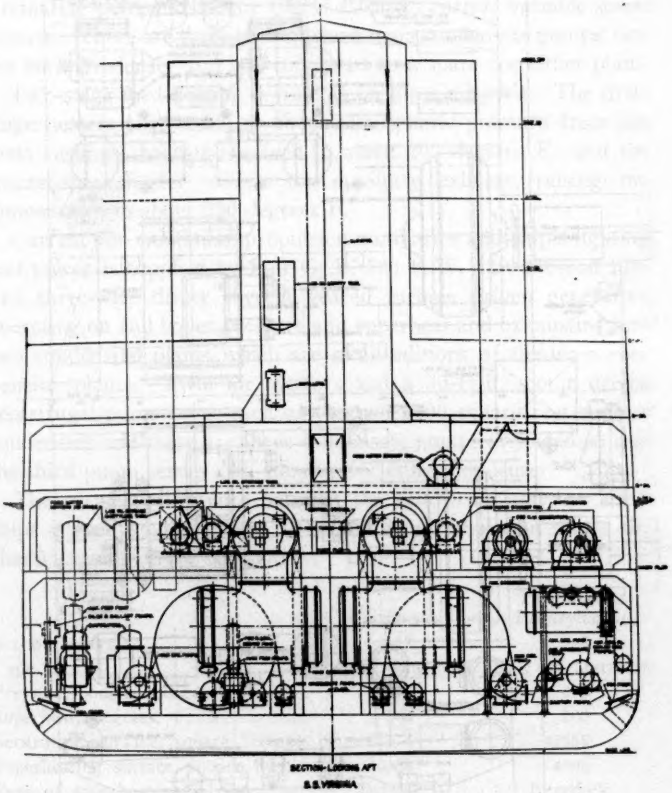
The greatest difference between the *California* and her sister ships is in the boiler plant. The following tabulation shows the characteristics of the two plants:

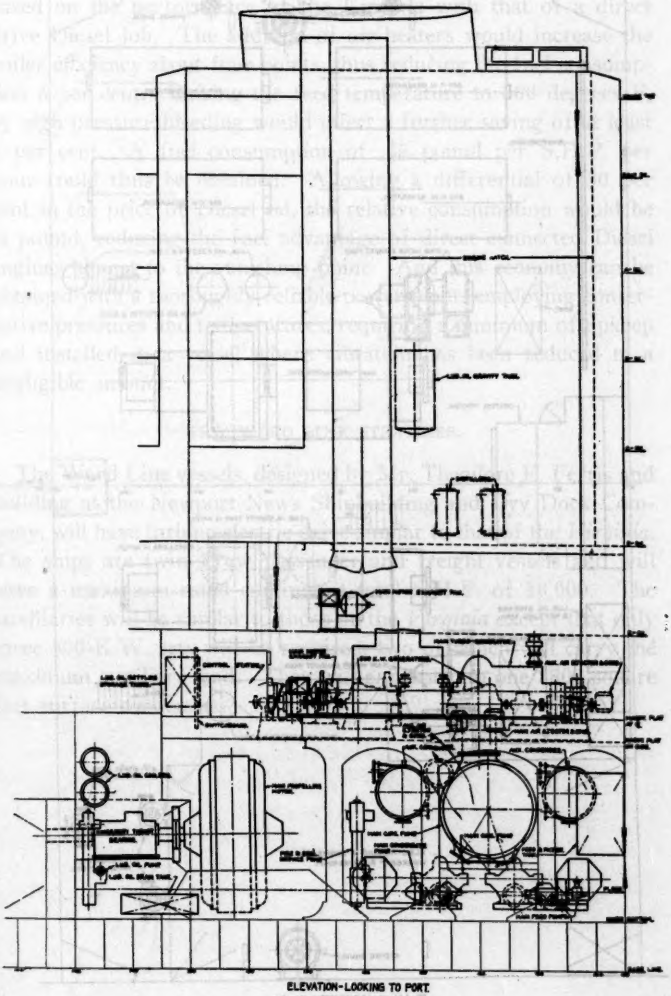
	California	*Virginia and Pennsylvania*
No. of boilers	12	8
Type	B. & W. Merchant	B. & W. Merchant
Pressure, pounds, gauge	275	300
Superheat, degrees, F.	100	200
Generating surface, square feet	55176	43688
Superheating surface, square feet	16128	4808
Type of superheater	Outside	Interdeck
Airheaters	None	None
Average fuel consumption in service, all purposes, pounds per S.H.P. per hour	.764	.723

It will be noted that the increase in pressure and superheat has produced a saving of about 5.5 per cent.

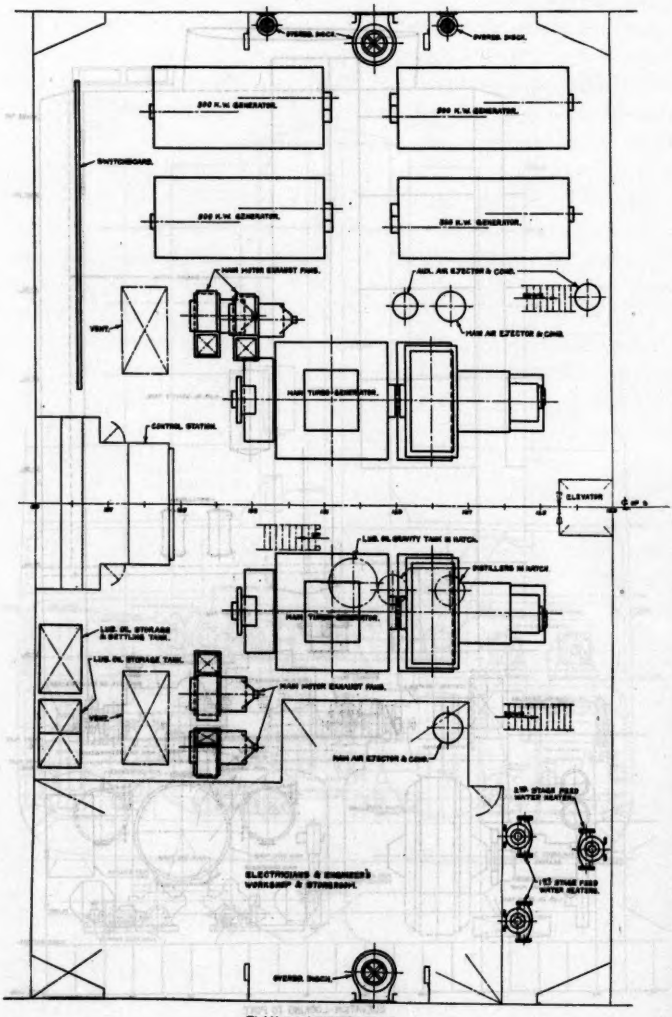


PLAN OF LOWER ENGINE ROOM.
S.S. VIRGINIA.


SECTION LOOKING AFT
S.S. VIRGINIA



ELEVATION—LOOKING TO PORT.
S.S. VIRGINIA.



PLAN OF MACHINERY FLAT
S.S. VIRGINIA.

Let us compare the economy of a turbine electric installation based on the performance of the *Virginia* with that of a direct drive Diesel job. The addition of air heaters would increase the boiler efficiency about four points, thus reducing the fuel consumption 5 per cent. Raising the feed temperature to 300 degrees F. by high pressure bleeding would effect a further saving of at least 1 per cent. A fuel consumption of .65 pound per S.H.P. per hour could thus be obtained. Allowing a differential of 30 per cent in the price of Diesel oil, the relative consumption would be .5 pound, reducing the fuel advantage of direct connected Diesel engines almost to the vanishing point. And this economy can be obtained with a thoroughly reliable power plant, employing conservative pressures and temperatures, requiring a minimum of upkeep and installed in a vessel where vibration has been reduced to a negligible amount.

NEW WARD LINE STEAMERS.

The Ward Line vessels, designed by Mr. Theodore E. Ferris and building at the Newport News Shipbuilding and Dry Dock Company, will have turbine electric drive similar to that of the *Virginia*. The ships are twin-screw passenger and freight vessels and will have a maximum rated continuous total S.H.P. of 16,000. The auxiliaries will be similar to those on the *Virginia* except that only three 500-K.W. sets will be required, two of which will carry the maximum auxiliary load. They will exhaust into one 2800 square feet surface condenser.

NOTES.

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3. H. M. Battleships "Nelson" and "Rodney."
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4. Sea Trials of Italian Destroyers.
Shipbuilding and Shipping Record, March 28, 1929.

5. Ricardo-Brotherhood High-Speed Diesel Engine.
Marine Engineer and Motorship Builder, March, 1929.

6. 750 B.H.P. Vickers High-Speed Oil Engine.
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7. The Wear of Diesel Cylinder Liners.
Marine Engineer and Motorship Builder, March, 1929.

8. The Impractical Man.
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10. Oil Separating Barges for the French Navy.
Shipbuilding and Shipping Record, March 7, 1929.

11. The Navy Estimates (British) for 1929.
The Shipbuilder, April, 1929.

12. Electron.
L'Ala d'Italia, March, 1929.

WHAT IS THE MOST ECONOMICAL DRIVE FOR SHIPS?

By C. R. Waller, Chief Engineer De Laval Steam Turbine Co.

Approximately 60 per cent of the gross tonnage of new vessels launched in 1927 were propelled by steam, and of the steam tonnage, slightly over 70 per cent were propelled by reciprocating engines, the remainder by turbines. Of the new ships launched during the past three years, the annual tonnage of reciprocating engined ships has declined 55 per cent, while steam turbines have increased 48 per cent and oil engines 30 per cent.

It has been claimed that fuel costs with Diesel engines are lower than with steam drive, but in order that this may be true at present prices for Diesel engine oil and for bunker oil for burning under boilers, the Diesel engine can be allowed not more than about half as much oil per shaft horsepower (not indicated horsepower) as is required by the steam turbine. Actually, the Diesel engine requires around .42 pound per shaft horsepower for engine only, and .48 pound for all purposes, of oil costing about $2.00 per barrel in New York, as against a cost of bunker oil for steam generation of about $1.05 per barrel. In addition, the Diesel engine consumes large quantities of high cost lubricating oil, while the turbine requires very little. The Diesel engine is also at a disadvantage because of higher interest, repair and depreciation charges and weight. In spite of the tremendous sums spent in converting Shipping Board vessels to Diesel drive, most marine engineers agree that Diesel engines are not at the present time to be considered for the average American merchant vessel, at least not for horsepowers of 2000 and over, except possibly for long voyages, as in the case of certain classes of tramps.

Recent improvements in equipment put steam on a new footing. These innovations include higher steam pressure, higher superheat, higher vacuum and the regenerative heating of feed water by means of steam withdrawn from successive stages of the turbine. By virtue of these improvements, a shaft horsepower can be developed on a total fuel consumption for all purposes, including auxiliaries, of .6 to .65 pound of oil, or less than one pound of coal, and even better figures may be expected. Besides requiring less expenditure for fuel, the steam turbine power plant, particularly with mechanical reduction gear, weighs less and, in the larger powers, occupies less space, than does either a reciprocating steam engine power plant or a Diesel engine, and, furthermore, it is more reliable and easier to handle.

It thus appears that for power plants approaching in size to 4000 horsepower or over, the steam turbine is supreme, the only question remaining being the type of turbine, or rather, the type of power transmission between the turbine shaft and the propeller shaft, that is, whether mechanical speed reducing gears shall be used, or electrical generators, with switches, cables and motors. Much effort was at one time expended in the development of hydraulic speed reducers, but such have now been discarded because of inherently low efficiency.

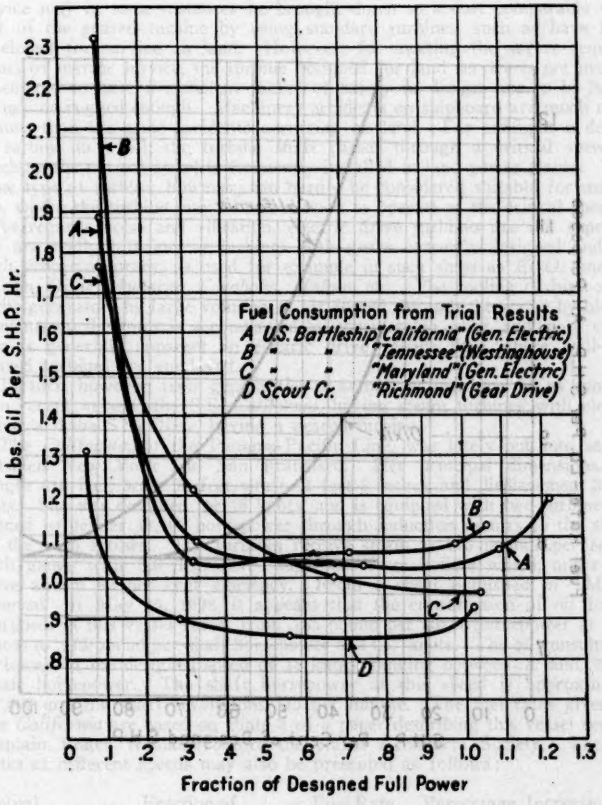
The case for electric drive has been ably, fully and repeatedly presented in magazine articles, newspaper interviews, and addresses before engineering societies. Claims are made of complete absence of noise and vibration, also of offering duplicate power units where more than one turbine is installed, and therefore greater safety and convenience, and finally of saving fuel. While the efficiency of transmission from turbine shaft to propeller shaft is admittedly not so high as with mechanical gears, due to losses in both generator and motor, it has been claimed that this is offset by higher turbine efficiency consequent upon running at approximately the designed turbine speed at all times, instead of according to the actual speed of the ship.

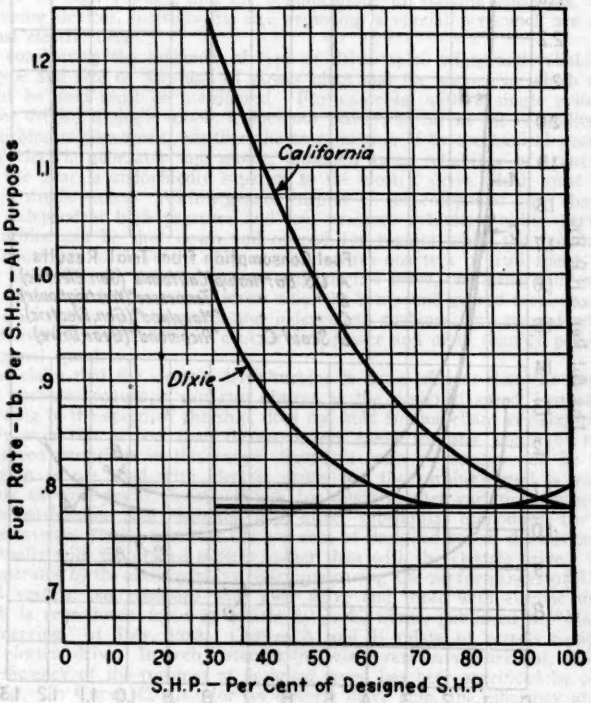
The prima facie case against electric drive is, on the other hand, fairly obvious, since the interposition of generators, switch gear, cables, controllers and motors between turbines and propeller shafts involves increased cost, increased losses (4 to 7 per cent greater), increased space and weight, and increased complication and attendance, as compared with the use of simple mechanical speed reducers. Electric generators, control equipment, wiring and motors are more intricate than are mechanical gears and require more attention. The liability to short-circuit because of salt water dampness, the dangers of high voltage, and the complications of starting rheostats, synchronizing devices, instruments, etc., requiring a special personnel, are also against electric drive.

In considering the influence of type of drive upon safety and reliability, the type and size of ship and of power plant and the service in which they are to be used must be considered. For example, with a single cylinder turbine driving a single screw, it does not matter, so far as the reliability of the turbine is concerned, whether the transmission is by mechanical gearing or by electric generator and motor, although as to reliability of transmission, the gear is undoubtedly superior to the electric drive which must rely upon a single motor. With a geared turbine of the compound type, that is, with independent high pressure and low pressure cylinders, either part of the turbine can be shut down and opened for repairs while the remaining part continues to drive the ship at about three-quarters of full speed. In order to realize the same advantage with electric drive, two generating units must be installed. On a twin screw ship, the compound geared turbines constitute in reality four separate and independent turbines, any two of which can deliver more than 50 per cent of full power and more than 75 per cent of normal speed.

The claim that the electric drive turbine is more efficient than the geared drive turbine because it can run always at the designed speed, instead of according to the speed of the ship, does not hold for merchant vessels, which hardly ever run at less than three-quarters speed for any length of time. For speed variations in this range change in ratio of transmission by pole changing is not used with electric drive, but the turbine speed is varied exactly as with geared drive. While for much greater variations in speed, as on battleships, pole changing may effect substantial economies, for the range between 75 per cent and 110 per cent of designed speed, the advantage is actually with the geared turbine rather than with the electric drive. This is illustrated by the accompanying chart comparing the performances of U. S. naval vessels, one equipped with gear drive and three with electric drive, which is reproduced from an article by J. C. Shaw, published in "Marine Engineering" of July, 1923. Curves A and B relate to vessels equipped with electric drive. In both instances the water rate curves are flat, because the efficiency of the turbines at designed speed has been sacrificed by overloading. In Curve C, also for an electric drive ship, the efficiency at designed speed has not been sacrificed, but the curve slope is steeper, due to the fact that a turbine with a comparatively small number of rows of blades was used. Curve D relates to a ship with geared turbines and shows not only a lower water rate than is obtained with the electric drive ships, but also a flatter curve. The designed power for a merchant vessel would be at about .8 on the abscissa, of this chart, and three-quarters speed would correspond to about .4, giving a uniformly good efficiency for the entire range of normal operation.

Where a lower speed is needed for prolonged periods, as in the case of naval vessels, a comparatively inexpensive high pressure cruising turbine can be added to the geared turbine, as has been done on American scout


Fuel Consumption from Trial Results
A U.S. Battleship "California" (Gen. Electric)
B " " "Tennessee" (Westinghouse)
C " " "Maryland" (Gen. Electric)
D Scout Cr. "Richmond" (Gear Drive)
Lbs. Oil Per S.H.P. Hr.
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1.0
.9
.8
A
B
C
D
0 .1 .2 .3 .4 .5 .6 .7 .8 .9 1.0 1.1 1.2 1.3
Fraction of Designed Full Power


1.2
1.1
1.0
.9
.8
.7
California
Dixie
Fuel Rate - Lb. Per S.H.P. - All Purposes
0 10 20 30 40 50 60 70 80 90 100
S.H.P. — Per Cent of Designed S.H.P.

cruisers. By this means good efficiencies can be shown by geared turbines at cruising speeds, but with less added weight, cost and complication than would be involved in electric drive. The use of a cruising turbine gives in effect a turbine with an increased number of stages, which offsets, or more than offsets, the lower speed of revolution of the main part of the turbine.

First cost has also to be considered. The cost of electric drive for marine service may in some instances be brought down to a cost comparable with that of the geared turbine by using standard turbines, such as have been developed for service on land. However, for meeting the severe requirements of marine service, the turbine designed for land service is not usually considered to have a sufficient factor of safety in design, nor to be heavy enough or rugged enough. Machinery accidents on shipboard are much more serious than the same accidents occurring on land. For example, a design of turbine in which the turbine shaft passes through a critical speed in reaching the normal speed is frequently installed in land power plants. This same type of turbine, however, can hardly be considered suitable for marine use, where the turbine may be called upon to operate at the critical speed.

As regards noise and vibration, electric drive turbines are not superior, but are really inferior, to turbines with gears correctly designed and cut with proper accuracy, as used for example in such ships as *El Oceano, El Coston, Dixie, Munargo, Carabobo, Malolo,* etc. The cooling of high-speed turbo-generators by large volumes of air forced through the stator by blower elements on the rotor is accompanied by considerable noise, and more vibration is generally apparent on electric drive vessels than where well-built geared turbines are employed.

To turn, however, from generalities to actual performance, let us consider two recent vessels, the S.S. *California,* having steam turbines with electric drive, and the S.S. *Dixie,* having a geared turbine.

The *California,* of the Panama-Pacific Line, was lately put into service between New York and San Francisco. Her principal dimensions are, length 610 feet, beam 80 feet, draft 32 feet 3 inches, and displacement 30,250 tons. She was designed for 18 knots, and is equipped with two turbines designed to deliver 17,400 horsepower through induction motors to the shafts of the twin screws. The turbines receive steam at 275 pounds per square inch gauge with 110 degrees F. superheat from boilers which, under test, have shown 82 per cent efficiency. From a chart published in "Marine Journal" of May 15, 1928, it appears that the consumption of oil for all purposes in this vessel varies from .785 pound per shaft horsepower at 18.45 knots to .975 pound per shaft horsepower at 14.57 knots. The oil consumption is lowest at the designed speed of 18 knots, ranging between .77 and .79 per shaft horsepower. The shaft horsepower at this speed is approximately 17,000 and the shaft revolutions 115 per minute. The fuel rates given for the *California* are based on Plate 3 of a paper describing this vessel read by Captain Roger Williams before the Naval Architects Society. The fuel rates at different speeds may also be presented as follows:

Actual Speed	Fraction of Designed Speed	Fuel Rate Lb.	Percentage Increase Over Rate at Designed Speed
18	1.0	.78	0
16.2	.9	.85	9
14.4	.8	.98	26
12.6	.7	1.18	51

For comparison, we have the much smaller *Dixie,* built by the Federal Shipbuilding & Dry Dock Company for the Southern Pacific Steamship Lines (Morgan Line), and operating between New York and New Orleans. The

dimensions of the *Dixie* are, length overall 445 feet, length between perpendiculars 427 feet, beam 60 feet, draft 26 feet, and gross tonnage 8188. She is fitted with a De Laval compound turbine with double reduction gears designed to develop 7100 normal shaft horsepower (with 10 per cent maximum overload capacity) at 90 revolutions per minute propeller speed, and is capable of a speed of 16 knots. The turbine is designed to receive steam from B. & W. boilers at 330 pounds gauge and 200 degrees F. superheat. Air heaters recover heat from the stack gases. The auxiliaries are steam driven, including two generator sets supplying current for lighting and cooking, the exhaust of which is used for heating feed water, for heating the vessel and for hot water service to all staterooms. The cargo capacity is 5125 long tons, or 414,055 cubic feet (6000 cubic feet of which is baggage space), the machinery occupying a space of 71½ feet fore and aft.

The official trial of the *Dixie* was conducted during voyage No. 2 and the following data were taken from the official log covering this trial run. Four boilers were used most of the time, but on a number of occasions, for periods of about 24 hours each, one boiler was cut out and the ship run on three. At the higher rate of driving the temperatures in uptakes, air heater boxes, etc., were raised, resulting in .6 per cent lower boiler efficiency, but increasing the superheat so that the turbine efficiency was improved about 3 per cent. The auxiliaries were operated on boiler steam at reduced pressure, exhausting under back pressure to the heating system, feed water heater and the low pressure section of the main turbine, except during certain periods of the Southbound and Northbound trips, when the exhaust of all auxiliaries was led into the auxiliary condenser in order to determine the water rate of the main turbine without auxiliaries. Otherwise, the methods of operation were normal.

The builder's trial trip covered a complete round trip from New York to New Orleans and back to New York, and the following are averages of the readings taken during this trip:

	Southbound	Northbound
Mean displacement, tons	10,220	11,830
Mean draft	22 ft. 1½ ins.	24 ft. 11¼ ins.
Distance run, miles	1607	1609
Time elapsed	105 hr. 39 min.	95 hr. 40 min.
Mean speed, knots	15.17	16.81
Mean R.P.M.	90.35	90.28
Mean shaft horsepower	7600	7535
Mean fuel consumption, pound oil per S.H.P.	.783	.7975
Heat content of oil, B.T.U. per pound	18,190	18,010
Mean steam pressure, pounds	335.5	332.2
Mean steam temperature at turbine, F.	602.7	603.6
Mean vacuum, corrected, inches	28.18	28.44
Back pressure on auxiliaries, pounds	12.9	14.54
Mean auxiliary generator load, K.W.	64.9	61.1.

During the Southbound trip a six-hour fuel and water rate test was run at 14.4 knots. The fuel consumption was determined by tank soundings as well as by meter readings, and the steam used by all of the auxiliaries was condensed in the auxiliary condenser and measured. The overall fuel rate per shaft horsepower was .764 pound, the auxiliaries using 25 per cent of the total steam, as determined by measurement. In a similar twelve-hour test

OPERATING RESULTS OF THE "DIXIE."

Voy. No.	R.P.M. South-bound	R.P.M. North-bound	SPEED (Knots) South-bound	SPEED (Knots) North-bound	Fuel Oil Gals. South	Fuel Oil Be.
1	75.3	79.7	13.0	14.8	61300	13.72
*2	90.5	90.3	14.8	16.7	87500	13.45
3	74.3	79.0	12.3	15.0	59600	13.52
4	76.0	75.5	12.4	14.9	59400	16.18
5	72.7	79.6	12.4	15.0	53300	16.45
6	71.9	77.9	12.1	15.1	53000	16.85
7	70.7	82.2	12.2	15.3	53000	16.30
8	71.0	78.0	12.3	14.9	55000	15.48

Voy. No.	Fuel Oil Gals. North	Fuel Oil Be.	** Mean Draft	*** Mean Dplt. Ton	Lbs. Oil SB + NB	Time Hrs.
1	60790	13.45	22-6-	10475	991500	247.12
*2	79220	13.52	23-0½	10750	1356000	218.12
3	57500	16.18	23-0½	10750	943000	253.6
4	52500	16.45	21-8-Æ	10000	893000	253.55
5	58700	16.85	22-6½	10460	921000	249.33
6	55300	16.30	22-5-Æ	10400	862000	254.28
7	60500	15.48	22-9-¾	10620	808000	254.25
8	56000	13.25	22-5-¾	10420	897000	253.07

NOTE:—* Trial trip. ** Full load draft, 26 ft. 0 in. *** Full load displacement, 12440 tons.

on the Northbound trip, the following values were obtained: Speed, 16.71 knots; overall oil consumption, .780 pound per shaft horsepower; steam used by auxiliaries, 25.05 per cent.

Four boilers were in use during these several tests, the superheat ranging from 170.8 degrees F. to 178.2 degrees F., as against the designed value of 200 degrees F., and the pressure at the boilers varied from 344 to 344.1 pounds as against the designed value of 350 pounds, while the corrected vacuum was almost exactly 28 inches, as against the designed value of 28.5 inches.

To place the comparison with the *California* on more nearly the same footing, assume the auxiliaries of the *Dixie* to be electrically driven from a generator on the main turbine shaft, as are those of the *California*, and the feed water to be heated by steam bled from the main turbine, the power plant otherwise remaining unchanged. This gives the following figures:

Dixie with *steam auxiliaries:* .783 pound of oil for all purposes per shaft horsepower. *Dixie* with *electric auxiliaries:* .66 pound of oil for all purposes per shaft horsepower.

California with *electric auxiliaries:* .77 pound of oil for all purposes per shaft horsepower.

The power plant of a new vessel designed along the lines of the *Dixie* could be still further improved. For instance, modern high efficiency water tube boilers would have an efficiency of about 85 per cent, including air heaters or economizers, while high efficiency, cross compound turbines with double reduction gears receiving steam at 400 pounds gauge and at 725 degrees F., and exhausting to 28½ inch vacuum, would have a practically flat efficiency curve of about 75 per cent over the working range, giving a fuel

consumption of about .6 pound of oil per shaft horsepower for all purposes. The generator supplying power to electric driven auxiliaries would normally be coupled directly to the main turbine, but would also be arranged to be driven by a standby turbine or internal combustion motor for use when the main turbine was turning at slow speed, or shut down altogether, as at dock. The motor driven condenser circulating pumps could be arranged to draw from the hold in case of necessity, supplementing the bilge pumps.

The recorded operating results of the *Dixie* for the first eight round trips between New York and New Orleans are given in the above table.

It will be noted that the mean speeds made on individual trips to suit the running schedule vary from 12.1 knots to 16.7 knots, 16 knots being the designed speed, and are thus within the range of high turbine efficiency, even though the speed of the turbine does bear a fixed ratio to the speed of the ship.

DISCUSSION.—BY CAPTAIN S. M. ROBINSON, U. S. N.

What is the most economical drive for ships? It seems to me that the answer to this is very simple. There isn't any such thing. This is entirely too broad a subject to cover with a single method of propulsion.

Of all the methods discussed by the author (Mr. G. R. Waller) in his paper (published in Feb. 1 and Feb. 15 issues of "Marine Journal"), I think it can be readily demonstrated that any one of them is the most economical for some type of ship performing some particular kind of duty, and it seems to me that the only rational method of discussing such a question is, first, to define the kind of ship, and, second, to give the discussion. No better example of the futility of a general discussion could be given than the set of curves which the author furnishes to show the relative superiority of the light cruiser *Richmond* over the battleship *California*. He might just as well have compared a barge with a racing motor boat.

DIFFERENCES OF OPINION.

Now I don't believe the most enthusiastic "electric driver" would propose this form of propulsion for the *Richmond*. There are certain ships for which geared turbines are so obviously the right thing that most people don't waste any time discussing them. The *Richmond* turbines have speed reductions of 4 and 6.8 for the low pressure and high pressure turbines, respectively; why mention anything else? There are other classes of ships for which the best type of propulsion will always be a subject of argument, with both sides well fortified with good data, and there are still other classes for which electric propulsion seems ideal, to say nothing of the other classes for which neither of these methods is as good as some of the forms of propulsion which the author of this paper has relegated to the scrap heap.

BATTLE OF THE POWERS.

But to consider some of the details of this paper. The first part of the paper deals with the Diesel engine. At the present time the whole subject of fuel costs is in a state of flux; powdered coal and stoker-fired boilers are being considered seriously. It is true that at the present time there is a great disparity between the prices of bunker oil and Diesel oil, but if one stops to consider that for several years past the Diesel engine has had a tremendous advantage over the steam turbine, even at the prevailing prices of

oil, then it is easy to see why the present condition exists. The steam turbine, fighting for existence, has forged far ahead. Is it reasonable to believe that the Diesel engine will stand still and allow present conditions to continue? I think not.

One does not have to be much of a prophet to foresee improvements in the Diesel engine which will enable it to use lower grades of fuel. Certainly, this would be no more difficult to imagine than was the use of powdered coal on board ship a few years ago. In other words, the old war between the Diesel engine and the steam turbine is not over yet by any means, and probably will not be until some very radical change in the fuel situation takes place.

GEAR VS. ELECTRIC DRIVE.

In the discussion of gears versus electric drive, we find the following statement: "The prima facie case against electric drive is, on the other hand, fairly obvious, since the interposition of generators, switch gear, cables, controllers and motors between turbines and propeller shafts involves increased cost, increased losses (4 to 7 per cent greater), increased space and weight, and increased complication and attendance, as compared with the use of simple mechanical speed reducers. Electric generators, control equipment, wiring and motors are more intricate than are mechanical gears and require more attention. The liability to short-circuit because of salt water dampness, the dangers of high voltage, and the complications of starting rheostats, synchronizing devices, instruments, etc., requiring a special personnel, are also against electric drive." Statements of this character were quite common ten or fifteen years ago, but I thought that all engineers had abandoned them by this time. They were not so unreasonable when discussing something that existed only on paper or in the brain of some engineer, but the past performance of electric drive ships has completely disproved such statements. Having followed the operation and repairs of machinery of both these types of propulsion for quite some years, I can assure the author that he need no longer worry over these "defects" of electric drive.

RELIABILITY AND EFFICIENCY.

The next paragraph of the paper deals with comparative reliability. That seems to me to be a complete waste of time. From an engineering point of view, they are both thoroughly reliable. For military purposes, entirely new arguments may come up, but they have nothing to do with reliability "per se" which is thoroughly established by both forms of propulsion.

The next paragraph states, "The claim that the electric drive turbine is more efficient than the geared drive turbine because it can run always at the designed speed, instead of according to the speed of the ship, does not hold for merchant vessels, which hardly ever run at less than three-quarters speed for any length of time." I have never heard anyone make this claim for electric drive. Speed changes are obtained in exactly the same way they are with the geared turbine, to wit, by changing the turbine speed. The advantage of being able to have a variable speed reduction with the electric drive (by pole-changing) is, however, a very great advantage for some types of ships and should not be minimized.

There are a few other statements that should be noted. For example, the author says, "As regards noise and vibration, electric driven turbines are not superior, but are really inferior, to turbines with gears correctly designed and cut with proper accuracy, as used, for example, in such ships as *El Oceano, El Coston, Dixie, Munargo, Sarabobo, Malolo,* etc. The cooling of high speed, turbo-generators by large volumes of air forced through the stator by

blower elements on the rotor is accompanied by considerable noise, and more vibration is generally apparent on electric drive vessels than where well-built geared turbines are employed." Surely the author has forgotten the experience of some of our ships with reduction gears where special boards were appointed to find a method of reducing the noise of the gears. Now the truth of the matter is that good installations of either kind are satisfactory in this respect, but there is unquestionably some advantage, in this respect, for electric drive. The author should stand alongside one of the *Lexington's* main generators before condemning the electric drive for being noisier than gears.

In the second part of the paper (appearing in "Marine Journal" Feb. 15), we get down to what appears to be the real purpose of the paper, and that is a discussion of what is the best form of propulsion for high speed or fairly high speed passenger ships. Shall it be geared turbines or electric drive? That is something more definite than ship propulsion. The treatment in the paper, unfortunately, branches off again into a comparison of wholly dissimilar ships. It is true that the author recognizes this fact in the last part of his paper and makes the following statement: "To place the comparison with the *California* on more nearly the same footing, assume the auxiliaries of the *Dixie* to be electrically-driven from a generator on the main turbine shaft, as are those of the *California*, and the feed water to be heated by steam bled from the main turbine, the power plant otherwise remaining unchanged." Well, to begin with, the auxiliaries of the *California* are not driven in that fashion. Also, in the very laudable desire to reduce these ships to a common basis, why does the author hit on a supposed difference which will boost the showing of the geared drive when there are many more differences between the two ships, all of which would reduce the *California's* fuel consumption. Some of these are the higher steam pressure and superheat used in the *Dixie's* turbines.

A set of curves of performance is given for the *Dixie* and *California*. The disparity between these two curves could be made much greater if the low point of the *Dixie's* curve were placed at 70 per cent instead of 80 per cent as the author has done, but why not place the low point of each at 100 per cent; then we would have a true comparison of the two turbines and we should also find that the two curves are close to coincidence. The author's comparison of the performance of the *Dixie* and *California* tends to show, if it shows anything at all, that there is no great difference between the economy at designed speed, of the electric drive, and the *double reduction gear*. We are now down to the real crux of this problem, and that is the fact that the geared turbine completely loses all of its advantages over the electric drive as soon as the reduction is so great as to make a double reduction necessary.

In studying this problem, there is another point that should not be lost sight of by marine engineers, shipowners, and others interested in the future of our merchant marine. In the first place, a thorough study of the "Jones-White" bill is in order. In the second place, it should be recognized that high speed passenger vessels are, in effect, auxiliary naval vessels, and all legislation which has been passed for the assistance of the merchant marine takes this into account. The higher the speed of these vessels, the better auxiliaries they will make; at the same time, they have got to be operated so as to show a profit as merchant vessels and this may require operation at a much lower speed than is desired for naval purposes. It seems to me that this is one of those cases where electric drive will prove to be absolutely the most satisfactory solution of this problem to both shipowner and the government which is backing him.

REPLY.—BY C. R. WALLER, CHIEF ENGINEER DE LAVAL STEAM TURBINE CO.

In your issue of March 15 appears a discussion by Captain S. M. Robinson of my article entitled as above. I wish to take this opportunity to express appreciation of his having entered into this discussion, as I am well aware of his wide experience, particularly with naval vessels. His remarks may be classified under the following heads:

1. The fairness of my comparisons of vessels using gear transmission with vessels using electric transmission.
2. The status of the Diesel engine as a competitor of steam.
3. Relative cost, space, weight, complication, attendance, reliability, vibration and noise as between electric and gear drives.
4. Efficiency comparisons, including efficiency at reduced speeds, and as affected by steam conditions; and
5. Bearing of the Jones-White bill, that is, of the possibility of using high speed passenger and cargo ships as naval auxiliaries.

Before tackling the gear vs. electric issue, it may be well first to dispose of the Diesel engine question. Captain Robinson says that the whole subject of fuel costs is in a state of flux, and asks whether it is reasonable to believe that the Diesel engine will stand still and allow present conditions to continue. He thinks not, but offers no suggestion as to what future developments may be, leaving the matter purely speculative. I do not pose as a prophet, either. My statements as to the relative economies of Diesel engines and modern steam turbine power plants were based upon present-day facts, the facts which must be considered by the private shipowner who is about to lay down his money for a new vessel, and who is interested primarily in keeping his operating costs, including the cost of fuel, as low as possible.

As to the fairness of comparisons, the steam turbine was compared with a Diesel engine "for horsepowers of 2000 and over" and electric drive with geared drive "for power plants approaching in size to 4000 horsepower and over." The two vessels principally compared, namely, the *California* and the *Dixie*, are both high speed passenger vessels, although the *California* has about two and a half times the displacement and rated horsepower of the *Dixie*, and other things being equal, the larger power plant would be expected to lend itself to higher efficiency.

Captain Robinson criticises the use of J. C. Shaw's diagram comparing the performance of the geared turbine scout cruiser *Richmond* with that of the electric driven battleship *California*. These curves were presented to illustrate respective characteristics of the two systems of propulsion, and I do not believe that re-reading of my article will disclose anything resembling the comparison of a barge with a motorboat. Mr. Shaw's position and experience qualified him to speak with authority and his figures, to the best of my knowledge, have never been challenged successfully. Moreover, the propelling equipment of the *Richmond* could be transferred to the battleship *California* with very little modification, except that the *California*, being a slower ship, would have a somewhat slower propeller speed, although not so much slower but that it could be handled by a single geared turbine of the same general character as that installed on the *Richmond*.

Captain Robinson advises that we should not worry about such questions as increased cost, increased space and weight, and increased complication and attendance of electric drive, as compared with gear drive. Shipowners must worry about these things, however, as they discover material differences when proposals for the two types of drive are compared. It cannot be denied that electric drive equipment is more intricate than speed reducing gears, nor that it weighs more and generally costs more. Nor is there any question, as

is admitted by Captain Robinson, that correctly designed and accurately cut gears run without noise or vibration. Such gears are obtainable on the market.

Now as to comparative operating costs and efficiencies. First, let me clarify a statement made in the original article to the effect that the losses with electric transmission are 4 to 7 per cent greater than with gear transmission; these percentages applied to the total power delivered by the turbine. In other words, the transmission losses with electric drive, including exciters and generator and motor ventilating equipment, are two to three times as great as with gear drive. The efficiency of transmission ranges from 98½ per cent with a single reduction gear to 97 per cent with a double reduction, while the combined efficiency of generator and motor will be between 90 and 95 per cent.

It is the comparison between the passenger vessel *Dixie* and the passenger vessel *California* that Captain Robinson selects as the real issue, as does also "Veritas," whose letter appears in your issue of March 1. Captain Robinson suggests that the disparity between the curves for the *Dixie* and *California* respectively, "could be made much greater if the low part of the *Dixie's* curve were placed at 70 per cent instead of 80 per cent, as the author has done, but why not place the low point of each at 100 per cent; then we would have a true comparison of the two turbines and we should also find that the two curves are close to coincidence." The answer is that the curves were not doctored to favor either vessel, but represent actual ratings of the propelling machinery of the two vessels and the maximum power point of one has been made to coincide with the maximum power point of the other. In other words, the 100 per cent designed horsepower in each case represents the maximum continuous load that the makers of the respective propelling equipments guaranteed according to the builder's specifications. The *California* curve cannot be extended and made to assume the same shape as the *Dixie* curve, as to do so would presumably overload the generators and motors, since, according to data already published, the normal full load rating of the generators was 5250 kilowats each, with a permissible maximum rating of 6600 kilowats. The latter corresponds to the 100 per cent turbine rating given in the curve. This maximum power is presumably used only in cases of emergency, such as making up lost time due to bad weather, and it is more desirable that maximum turbine efficiency be obtained at the usual running speed, as in the *Dixie*, rather than at the maximum power, as in the *California*. Any well-designed passenger ship running on a fixed schedule must have excess power to make up time unavoidably lost on account of weather conditions.

Captain Robinson says that, "the geared turbine completely loses all of its advantages over the electric drive as soon as the reduction is so great as to make a double reduction necessary."

I am at a loss to understand just what Captain Robinson means by this statement, as it is well known that for commercial vessels, the double geared turbine weighs less, costs less and is more efficient than the single geared turbine. The turbine of a double-geared unit can, for the same size and steam conditions and because of the more favorable speeds possible, be designed to be more efficient than the turbine for a single-gear unit. The higher turbine efficiency possible will more than offset the slightly lower gear efficiency. As to the high efficiency and entire reliability, and the freedom from noise and vibration of properly designed double reduction gears, the performances of such ships as the *Dixie, El Oceano, El Coston* and the *Munargo* bear ample evidence.

Finally we come to the naval auxiliary argument. Speaking of high speed passenger vessels, Captain Robinson says, "The higher the speed of these vessels, the better auxiliaries they will make; at the same time, they have got to be operated so as to show a profit as merchant vessels, and this may require operation at a much lower speed than is desired for naval purposes."

I have already pointed out in discussing the curves for the *California* and the *Dixie* that the maximum power given for the *California* is the highest power at which that vessel can be operated continuously, which means that from this point of view the *Dixie* is superior.

In an earlier article published in "Marine Journal" of April 1, 1928, Captain Robinson said: "Practically, it would probably work out best to speed up the turbine itself when using the vessel as a naval auxiliary, thus making it possible to get the best turbine design from the point of view of economy of fuel as a merchant vessel, and economy of weight of turbine as a naval vessel." This suggestion would of course apply equally to geared turbine as well as to electric drive.

To sum up, the advantages of the geared turbine in actual commercial competition with electric drive can be stated briefly as follows:

1. The cost of installation is less.
2. The weight of the equipment is less.
3. The fuel consumption is appreciably less, and this is true whether single or double geared units are considered. It also applies to running at reduced powers such as are ordinarily encountered in the operation of commercial vessels. The *cost* of fuel is also less as compared with Diesel engines.
4. The mechanical operation of a properly designed and constructed gear drive is as good as, or better than, that of any electric drive so far produced.—"Marine Journal," Feb. 1, Feb. 15, Mar. 15, and Apr. 1, 1929.

THE PROPULSION OF SHIPS BY MODERN STEAM MACHINERY.*

By J. Johnson.

Historical and other analogies have been invoked to show that steam propulsion has run its course and must now give way to more efficient machinery. Analogies, however, are sometimes constructed out of insufficient or unsuitable material, and in such cases may be dangerously misleading. It is the author's purpose to examine the more efficient forms of steam propulsion, and, incidentally, to compare them with the Diesel engine. By way of introduction, it may be remarked that those exclusively interested in the manufacture of steam machinery and boilers are naturally emphatic as to the virtues of such plant, while those specially interested in the development of the Diesel engine are eloquent in their claims for its supposedly superlative advantages. There probably has never been a period in the history of shipping when shipowners have been so vitally and intimately concerned with modes of propulsion, and it may be as well at the outset, therefore, to hazard the somewhat trite observation that the economics of ship operation constitute the final court of appeal in these matters. The problem is two-fold—engineering and fuel. The owner inclines strongly towards a low first cost; he also desires assurance of fuel supplies at a favorable price, and however sympathetically he may regard suggestions to use British fuel, he is obliged

* Paper read before the Institute of Naval Architects, March 21, 1929.

by the circumstances of his business to get the best value obtainable. Can the shipowner be expected to prescribe his requirements? He depends in a measure on the advice of the shipbuilder and others, but having regard to the diversity of opinion which exists in technical circles, and the difficulty of elucidating the problem, he is rarely presented with a clear issue. There is, therefore, pressing need for a critical comparison of the various forms of propulsion now engaging the attention of shipowners and shipbuilders. This, naturally, should be framed so as to include all terms pertinent to the inquiry, namely, reliability, first cost, fuel expenses and repair expenses (running and periodical).

Reliability is a major factor in all ships, increasing in importance with size and speed, until in the largest vessels it governs all other considerations. Whether the vessel be small or large, freedom from dislocation of service is being insisted upon in increasing degree by owners, who are well aware that fuel or other economies have to be seriously discounted if associated with interruptions in trading operations. Nevertheless, the demand for reduction in first cost of the vessel and for substantial economies in fuel is very insistent, and, in so far as these are attainable by technical advances in the propelling agent, they must receive serious consideration. Apart from small installations for which the triple- or quadruple-expansion reciprocating engine is specially suited, there are three types of steam plant, each with its variants, prominently in use and applicable over a wide range of tonnage. (*a*) The combination of reciprocating engine and low-pressure turbine. (*b*) The single-reduction geared turbine, using low-pressure superheated steam generated in Scotch boilers. (*c*) The single-reduction geared turbine, using high-pressure superheated steam generated in water-tube boilers. In order to illustrate these alternatives, comparison has been made of the performances of a number of passenger and cargo vessels engaged in the North Atlantic trade, representative of what might be described as the low-pressure moderate temperature steam system at an advanced stage of its evolution, and the high-pressure, high-temperature steam system at the initial stage of its development.

A study of the claims of steam plant, present or prospective, must necessarily be preceded by full consideration of the condenser problem. In this connection, the histories of the steamers *Melita's* and *Minnedosa's* main condensers are of especial interest as examples of acute tube deterioration. They are of the "Uniflux" type, with relatively high-velocity cooling water. The Admiralty mixture brass tube (7 Cu, 29 Zn, 1 Sn) had an average life of one year with or without an electrolytic process, but a few months of service produced failures sufficiently numerous to be an embarrassment under service conditions. In February, 1915, the *Minnedosa's* condensers were retubed with cupro-nickel tubes of 80 Cu, 20 Ni composition. At the same time, the *Melita's* condensers were tubed with alternate rows of 90 Cu, 10 Ni and 85 Cu, 15 Ni. There were no failures in either vessel during the ensuing year, and on examining a large batch from each condenser no sign of deterioration was found in the 80/20 tubes, and only the slightest suspicion of corrosion in the 85/15 tubes. The 90/10 tubes, however, exhibited the same appearance as the earlier brass tubes, practically the whole of the interior surfaces being lightly pitted. Although no perforation had occurred, it was clear that a comparatively short life only might be expected of these tubes. It was concluded, therefore, that anything less than 20 per cent of nickel was not sufficient, that 20 per cent might be adequate, but that an increase in the nickel content would probably be beneficial. The tubes in the *Melita's* condensers were replaced by 80/20, and a proportion of 70/30 (CuNi) tubes introduced, the latter being commercially available at this juncture.

During the second year there were no failures in the *Minnedosa's* condensers, and in January, 1927, after two years' service, a batch of tubes was withdrawn from both port and starboard condensers and subjected to careful scrutiny, when no indication of general deterioration was observed. During the third year there were four failures, and at the annual overhaul in December, 1927, 100 tubes were withdrawn from each condenser. These were very carefully examined, and while their general condition was good there were signs of slight corrosion in irregular patches, principally towards the ends, the depth varying from 0.004 inch to 0.010 inch. (Original thickness, 18 W.G.=0.048 inch.) During the fourth year there were two failures, and at the overhaul in December, 1928, batches of tubes were drawn and subjected to a microscopic examination. The following details were recorded and convey a fair idea of the general condition of the condensers.

Starboard Condenser.—Top nest, 48 tubes: 47 tubes good. 1 tube slightly corroded. Middle nest, 48 tubes: All good. Bottom nest, 17 tubes: 11 tubes good, 2 tubes partly corroded, patches about 0.010 inch deep, 4 tubes slightly corroded.

Port Condenser.—Top nest, 48 tubes: 24 tubes good, 21 tubes pitted at ends less than ½ inch along, 3 tubes pitted at ends. Middle nest, 47 tubes: 37 tubes good, 8 tubes slightly pitted, inlet end, 2 tubes pitted 0.015 inch deep, inlet end. Bottom nest, 17 tubes: 10 tubes good, 6 tubes slightly corroded, 0.003 inch, some one end only, 1 tube corroded 0.010 inch. In all there have been six tube failures in four years, and there is reason to believe that these had their origin in manufacturing defects. The most recent examination would seem to indicate that the life of these tubes will not be less than six years. There are fair grounds for supposing that the life of 70 Cu, 30 Ni tubes will be much longer in condensers designed on modern lines. In view of the severity of the case, the first two years' experience with the 80/20 tubes was thought to warrant the belief that a partial solution of the condenser problem was within reach. On the strength of this conviction, together with growing evidence in other quarters, it was decided that the question of adopting high-pressure water-tube boilers was ripe for examination. Practical aspects of the condenser question are dealt with later.

The performances of the steamers *Melita* and *Minnedosa* are a convenient starting point in the proposed review. These vessels are propelled by the well-known combination machinery consisting of four-cylinder reciprocating engines driving the wing screws, exhausting into a low-pressure turbine, actuating the center screw. Steam is supplied by five double-ended boilers, hand-fired, and fitted with Howden's forced draught. This type of machinery is installed in several important vessels, but, so far as the writer is aware, was, until recently, used only in association with saturated steam. In February, 1925, superheaters were installed in the *Minnedosa's* boilers and appropriate modifications made to her propelling machinery. A year later, the three-bladed built wing propellers were replaced by three-bladed solid ones of modern design, and the solid center propeller by one of modified dimensions, the proportions having been determined at the National Tank. The efficiencies of the new center and wing screws were investigated by the Tank authorities and found to be 55 per cent and 67 per cent, respectively. Table I on this page gives log abstracts from typical summer Atlantic passages.

The fuel rate per shaft horsepower/hour is derived from the combined indicated and shaft horsepowers, adjusted to a basis of shaft horsepower by assuming a mechanical efficiency of 92 per cent for the wing engines. As the boiler efficiency is of the order of 74 per cent, it is evident that an installation of this kind, using oil fuel and Scotch boilers having an efficiency of 82 per cent, might be expected to show a rate of 0.70 pound per shaft horse-

TABLE I.

—	Year.	Speed, knots.	Total S.H.P.	Tons of Coal per Day (Propulsion).	Lb. of Coal per S.H.P. per hour.		$\frac{D^{\frac{2}{3}} V^3}{S.H.P.}$	$\frac{D^{\frac{2}{3}} V^3}{\text{Tons per day (Propulsion).}}$
					Propulsion.	All Purposes.		
Saturated steam (original propellers)	1924	16·33	11,500	158	1·28	1·41	255	18,500
Superheated steam (150 deg. F.) (original propellers)	1925	16·53	11,650	137	1·09	1·22	253	21,500
Superheated steam (150 deg. F.) (new propellers)	1926	16·45	10,600	123	1·08	1·22	256	24,650
	1927	17·08	11,500	133	1·08	1·21	290	25,900

TABLE II.

—	Speed, knots.	S.H.P.	Daily Consumption (Propulsion).	Lb. Oil per S.H.P. per hour.		$\frac{D^{\frac{2}{3}} V^3}{S.H.P.}$	$\frac{D^{\frac{2}{3}} V^3}{\text{Tons per day (Propulsion).}}$
				Propulsion.	All purposes.		
			Tons.				
West	19·26	19,500	136	0·65	0·71	290	41,500
East	19·48	20,346	142	0·65	0·71	294	42,100

TABLE III.

Ship.	S.H.P. (On Service).	Daily Consumption of Oil. Tons (Propulsion).	Fuel per S.H.P.-Hour.		$\frac{D^{\frac{2}{3}} V^3}{S.H.P.}$	$\frac{D^{\frac{2}{3}} V^3}{\text{Tons day (Propulsion).}}$	Thermal Conditions.		
			Propulsion.	All Purposes.			Pressure at Engines.	Temperature at Engines.	Vacuum.
									In.
Empress of Canada ..	20,000	225	1·05	1·13	307	27,200	185	Saturated	28·5
Montclare	12,500	117	0·87	0·98	284	30,300	200	500 deg.	28·7
Minnedosa*	11,500	86·5	0·70	0·785	290	38,500	207	530 ,,	29
Empress of Australia	20,346	142	0·65	0·71	294	42,100	190	580 ,,	29
Duchess of Bedford ..	16,000	97	0·57	0·625	293	48,000	340	670 ,,	29
Duchess of Atholl ..	16,100	99	0·58	0·64	284	46,000	340	680 ,,	29

* Figures converted from coal to oil as explained in text.

Empress of Australia	20,346	142	0·65	0·72	293	43,000	340	670 „	29
Duchess of Bedford ..	16,000	97	0·57	0·625	293	46,000	340	680 „	29
Duchess of Atholl ..	16,100	99	0·58	0·64	284				

* Figures converted from coal to oil as explained in text.

TABLE IV.

Ship.	S.H.P. (Designed).	Total Weight of Propelling Machinery and Boilers (including Water).	Weight of Machinery, Boilers and Water per S.H.P.
		Tons.	Ton.
Empress of Canada ..	20,000	3,400	0·17
Montclare	12,500	2,411	0·193
Minnedosa	11,350	2,097	0·184
Empress of Australia ..	20,000	2,850	0·143
Duchess of Bedford	18,000	2,275	0·126

TABLE V.

——	Cost.	Pounds of Steam per Hour.	Evaporation per Square Foot Heating Surface.
	£		
Scotch boiler (complete ex shop)	8,100	33,350 (85 per cent. superheated. 15 per cent. saturated)	5·33
Working pressure, 220 lb. Heating surface (oil basis), 6,250 sq. ft. Smoke tube superheater, 620 deg. F. All mountings, lagging, superheaters, air heaters, furnace fronts, smoke-boxes and uptakes to funnel base.			
Water-tube boiler (complete ex shop)	6,900	27,800 (All superheated)	5·34
Working pressure, 350 lb. Heating surface, 5,200 sq. ft. Integral superheater, 680 deg. F. All mountings, lagging, superheaters, furnace fronts, and uptakes and air heater to funnel base.			

TABLE VI.

20,000 S.H.P. (Overload Power). 18,000 S.H.P. (Service Power).	Total Heating Surface.	Total Steam.	Evaporative Rating on Working Boilers. (Overload Power).	Cost.
	Sq. Ft.			£
6 Water-tube boilers Working pressure, 350 lb. Steam temperature, 680 deg. F.	31,200	167,000	5·35	41,400
1 S.E. Scotch boiler, saturated (working)	3,000	11,000	3·7	4,000
1 S.E. Scotch boiler, saturated (reserve)	3,000	—	—	4,000
				£49,400
6 D.E. Scotch boilers Working pressure, 220 lb. Steam temperature, 620 deg. F.	37,500	202,000	5·38	£48,600

power (propulsion); the coal coefficient 25,000 would thus become 38,500 for oil, a result rather better than is registered in the same service by somewhat similar twin-screw vessels built in 1922, and fitted with double-reduction turbines using steam superheated 150 degrees F. Interest in the combination type of machinery has recently been revived by the advent of the Bauer-Wach, Parsons, and other systems, which are thermally similar, but have the advantage of avoiding the use of a low efficiency center screw of high revolutions. In such installations of this power, the fuel coefficient, using oil, would probably be in the region of 40,000. An appreciable amount of oil is required for internal lubrication in reciprocating engines, and this cannot be removed with the degree of certainty essential for water-tube boilers. This precludes the use of such machinery in association with high-pressure steam.

A description of the *Empress of Australia's* machinery conversion has appeared in the technical Press,* but perhaps some further observations may opportunely be made here. The original water-tube boilers were extremely inefficient, of indifferent construction, and in a poor state of repair. Moreover, the vessel had about 1500 tons of permanent ballast. Having regard to all the circumstances, it was decided to adopt Scotch boilers of 220 pounds working pressure, fitted with smoke-tube superheaters capable of yielding a temperature of 620 degrees F. Considerable attention was given to the design of the new main and auxiliary machinery, boilers, and feed heating, but nothing of a novel character was introduced beyond two 165-Kw. Diesel generators, sufficient for ship's lighting and forced draught fans. Typical results registered on fair weather passages are given in Table II. The boiler efficiency maintained in service is 83 per cent; steam rate of the Parsons' turbines 8.30 pounds per shaft horsepower-hour (turbines only), and the feed temperature obtained by the use of auxiliary exhaust and low-pressure bled steam 230 degrees F.

The *Duchess of Bedford*,† commissioned in 1928, was the first of four passenger vessels built to the order of the Canadian Pacific Railway Company for their Liverpool-Montreal service, in which high-pressure steam and water-tube boilers are employed. The propelling installation comprises Parsons' turbines and Yarrow boilers. The major portion of the auxiliary machinery is motor-driven, energy being supplied by Diesel and turbo generators. Before examining in detail the issues involved by the use of high-pressure high-temperature steam, it is desirable to compare her service results with those of the earlier vessels. The boiler efficiency maintained in service is 85 per cent, steam rate 7.3 pounds (turbines only), and feed temperature 300 degrees F. In order to complete the comparison, in Table III results of the *Empress of Canada* and *Montclare* are included. These vessels, built in 1922, are propelled by double-reduction geared turbines, steam being supplied by Scotch boilers—pressure 215 pounds. The former uses saturated steam and the latter steam superheated to 550 degrees F.

Putting technical comparisons on one side for a moment, it is of interest to note that the *Duchess of Bedford* (gross tonnage, 20,000) has made a complete voyage Liverpool-Montreal-Liverpool on a total of 1443 tons of fuel, inclusive of port consumption, at an average speed of 17.8 knots, as compared with a voyage of the *Montclare* (gross tonnage, 16,314), under similar weather conditions, at a speed of 16.35 knots, and consumption of 2182 tons, or 739 tons less. Table IV shows the comparative weights of the installations.

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* See "Engineering," vol. cxxiii, pp. 733-772.

† See "Engineering," vol. cxxv, p. 675.

As the *Empress of Australia* may be considered representative of the low-pressure steam system at its best, it is instructive to compare the essential features of her installation, in terms of space, weight and fuel consumption, with those of the *Duchess of Bedford*. The figures show a reduction of 16 per cent in space, 12 per cent in weight, and 12 per cent in fuel rate in favor of the *Duchess of Bedford*. The *Empress of Canada* is now being re-engined, and having superheaters fitted to her boilers. The installation will be exactly similar in its essential features to that of the *Empress of Australia*, but the service power will be 26,000.

A new vessel, the *Empress of Japan*, in course of construction, will have a high-pressure installation of 30,000 shaft horsepower, working pressure 375 pounds, and steam temperature 700 degrees F. The consumption rates for all purposes for the *Empress of Canada* and *Empress of Japan* are expected to be 0.70 and 0.60 pound, respectively, and there is every reason to believe that these figures will be comfortably realized. This represents a saving in the fuel rate for the high-pressure system of 14 per cent. The saving in space and weight will be about 15 per cent and 12 per cent, respectively. The question of the relative costs of high-pressure high-temperature water-tube boiler plant and low-pressure moderate-temperature Scotch boiler plant calls for special remark. In order that a high efficiency may be attained with the conventional design of water-tube boiler, it is essential that the evaporative rating shall not exceed that adopted for Scotch boilers, with superheaters, namely, about 5.0 pounds to 5.5 pounds per square foot of generative surface. Allowing for the difference in steam rate, the water-tube boiler generative surface may therefore be 10 per cent to 15 per cent less.

The costs of water-tube and Scotch boilers are illustrated in Table V.

In high-powered passenger vessels, such as are exemplified in Tables VI, VII, VIII and IX, reserve boiler power is, of course, invariably provided. The comparisons in all cases have been made on the basis of an evaporative rating of about 5.4 pounds per square foot of generative surface, calculated on the overload power, which is in each case 10 per cent in excess of the service power. This also provides for the contingency of a fraction of the boilers not being immediately available on leaving port after a shortened stay or the shutting down of boilers in emergency at sea. Under any but abnormal circumstances, the evaporative rating at the service power would not exceed 6 pounds. All comparisons made between water-tube and Scotch boilers are in this sense on a perfectly comparable basis. For 18,000 shaft horsepower, using boilers of the type indicated in Table V, the high-pressure installation costs about the same as Scotch boilers, as indicated in Table VI.

When the *Duchess of Bedford* installation was projected, selection of the number and size of the boilers was influenced to a considerable extent by the somewhat hypothetical nature of the case. In the light of actual experience, the installation might quite prudently comprise four larger water-tube boilers and two smaller Scotch boilers, as shown in Table VII.

The high-pressure water-tube boiler does not, however, show to full advantage until larger units are employed, as illustrated in Tables VIII and IX. Apart, however, from the actual boiler costs, a pronounced simplification of steam-pipe systems and incidental fittings is secured, the full advantage of which only becomes apparent from a study of the completed detail designs (Figs. 1 and 2).

With high-pressure machinery, the condensing surface for a given shaft horsepower may be reckoned at 15 per cent less, due to lower steam rate and greater proportion of bled steam. The whole of the propelling and boiler-room auxiliaries are also correspondingly smaller. Notwithstanding the extra cost of special material for high-pressure turbine blading, the gross

TABLE VII.

20,000 S.H.P. (Overload Power). 18,000 S.H.P. (Service Power).	Total Heating Surface.	Total Steam.	Evaporative Rating on Working Boilers (Overload Power).	Cost.
	Sq. Ft.			£
4 Water-tube boilers Working pressure, 350 lb. Steam temperature, 680 deg. F.	31,200	167,000	5·35	39,000
1 S.E. Scotch boiler, saturated (working)	2,000	11,000	5·5	3,000
1 S.E. Scotch boiler, saturated (reserve)	2,000	—	—	3,000
				£45,000
6 D.E. Scotch boilers Working pressure, 220 lb. Steam temperature, 620 deg. F.	37,500	202,000	5·38	£48,600

Saving in favour of water-tube boilers = 7·5 per cent.

TABLE VIII.

33,000 S.H.P. (Overload Power). 30,000 S.H.P. (Service Power).	Total Heating Surface.	Total Steam.	Evaporative Rating on Working Boilers (Overload Power).	Cost.
	Sq. Ft			£
6 Water-tube boilers Working pressure, 350 lb. Steam temperature, 680 deg. F.	51,500	275,000	5·35	62,700
1 S.E. Scotch boiler Working pressure, 200 lb. (saturated)	3,300	16,000	4·85	4,370
1 S.E. Scotch boiler (saturated), reserve	3,300	—	—	4,370
				£71,440
10 D.E. Scotch boilers .. .. Working pressure, 220lb. Steam temperature, 620 deg. F.	62,500	332,000	5·32	£81,000

Saving in cost in favour of water-tube boilers = 11·5 per cent.

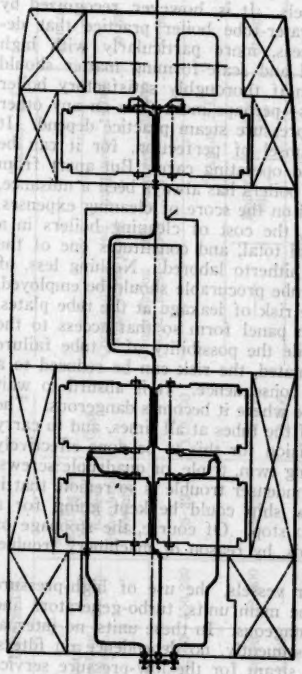


Fig 1. T.S.S. "EMPRESS OF JAPAN."
ARRANGEMENT OF STEAM PIPES IN BOILER-ROOMS.

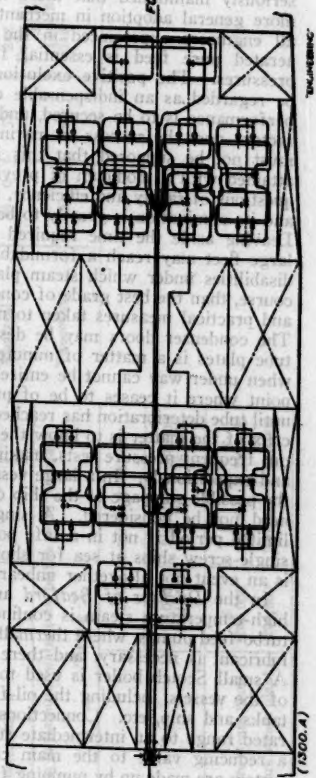


Fig 2. T.S.S. "EMPRESS OF CANADA."
ARRANGEMENT OF MAIN STEAM PIPES.

cost does not exceed that of the larger low-pressure set. The position is summarized in Table X. In both cases, electrically-driven auxiliaries have been allowed for, turbo and Diesel generators being included.

Having regard to the fact that water-tube boilers are used exclusively in the British and other navies, and in all modern power stations throughout the world, at pressures ranging from 250 pounds to 1300 pounds, it cannot be seriously maintained that there is today any insuperable obstacle to their more general adoption in mercantile vessels. It is, however, recognized by all engineers experienced in the best water-tube boiler practice that de-aerated pure feed is essential for success, more particularly with high pressures. The positive exclusion of oil and scale-forming matter should be regarded as an indispensable condition if thoroughly satisfactory boiler performance is to be secured, and on this—perhaps more than on any other factor—does the success of marine high-pressure steam practice depend. It must not be supposed that this is a counsel of perfection, for it can be attained with a modicum of prevision and operating care. But apart from questions of safety and efficiency, scale in boilers has always been a nuisance, and its elimination is greatly to be desired on the score of cleaning expenses. Leaving aside the time required in port, the cost of cleaning boilers in a large fleet may reach a formidable annual total, and constitutes one of the disabilities under which steam plant has hitherto labored. Nothing less, of course, than the best grade of condenser tube procurable should be employed, and practical measures taken to minimize risk of leakage at the tube plates. The condenser doors may be designed in panel form so that access to the tube plates is a matter of minutes. While the possibility of a tube failure when under way cannot be entirely eliminated, the risk can be reduced to a point where it ceases to be of practical consequence. It is absurd to wait until tube deterioration has reached a stage where it becomes dangerous. The crux of the matter is to know the state of the tubes at all times, and to carry out frequent pressure tests, making provision for this to be done effectively and expeditiously. In a large vessel having twin, triple, or quadruple screws, the possible stoppage of the ship due to condenser trouble is so remote that it need not be considered. A single-screw ship could be kept going for a limited period if not in a safe position to stop. Of course, the stoppage of single-screw ships at sea for short periods, by reason of machinery trouble, is an event not altogether unheard of.

In the *Duchess of Bedford* and sister vessels, the use of high-pressure high-temperature steam is confined to the main units, turbo-generators and turbo-feed pumps, where thermally advantageous. In these units, no internal lubricant is necessary, and there is, consequently, no dependence on filters. A small Scotch boiler is used to supply steam for the low-pressure service of the vessels, including the oil-fuel units, hydraulic-door pumps, heating of tanks and ship, etc. Connections are provided from this low-pressure saturated range to an intermediate stage of the main turbines, and also through a reducing valve to the main condensers. Feed-water losses in the main system are made up by pumping fresh water (or condensed evaporator steam) into the Scotch boilers, and opening the extra feed steam valves on the turbines. The dirty steam auxiliaries are definitely segregated from the high-pressure circuit. Those who have much experience of piping arrangements in ships know the extreme difficulty of preventing accidental contamination of fresh water by salt or oily water. The foregoing arrangement was therefore conceived as a practical means of obtaining the degree of security necessary in these installations, and, in fact, occupies a pivotal position in the general design. Its features, as established in actual service, may be summarized as follows:—(1) Absolutely clean boilers and superheaters—

TABLE IX.

66,000 S.H.P. (Overload Power). 60,000 S.H.P. (Service Power).	Total Heating Surface.	Total Steam.	Evaporative Rating on Working Boilers (Overload Power).	Cost.
	Sq. Ft.			£
9 Water-tube boilers Working pressure, 350 lb. Steam temperature, 680 deg. F.	103,500	550,200	5·32	119,700
1 S.E. Scotch boiler	3,650	21,000	5·75	4,500
1 S.E. Scotch boiler (reserve)	3,650	—	—	4,500
				£128,700
20 D.E. Scotch boilers .. .. Working pressure, 220 lb. Steam temperature, 620 deg. F.	125,000	660,000	5·28	£162,000

Saving in favour of water-tube boilers = 20 per cent.

TABLE X.

—	Total Cost of Installation.		
	18,000 S.H.P.	30,000 S.H.P.	60,000 S.H.P.
	£	£	£
Low-pressure steam plant ..	266,000	408,000	735,000
High-pressure steam plant..	256,000	390,000	690,000

TABLE XI.

—	Steam.	Diesel.
Consumption (all purposes)—18,000 shaft horse-power	0·63	0·48
Consumption (all purposes)—30,000 shaft horse-power	0·60	0·48

TABLE XII.

—	Boiler Oil.	Diesel Oil.	Steam.	Diesel.
			Daily Fuel costs for 18,000 S.H.P. (Atlantic Service)	
	s. d.	s.	£	£
New York ..	31 6	60	192	278
Montreal ..	43 0	70	262	324
			Daily Fuel costs for 30,000 S.H.P. (Pacific Service)	
Vancouver ..	36 0	46	348	355

the condition requisite for safety—absence of priming and sustained efficiency, enabling the boilers to be kept under steam for extended periods. (2) Reduction in boiler-cleaning costs. (3) The facility whereby substantial quantities of distilled water may be produced at sea or in port, either for make-up purposes or filling of boilers or reserve tanks. (4) An economical means of restoring the water losses in the main system. (There are other minor features which cannot be dealt with here, but which are designed to ensure ease of manipulation.)

Up to the present, the *Duchesses—Bedford* and *Atholl*—have each made six Atlantic voyages, a distance of 34,000 miles. During this period the boilers, propelling machinery (with the exception of a blade failure in an L.P. ahead turbine), and auxiliaries have been eminently satisfactory. Incidentally, an average speed considerably above the contract figure has been maintained. The condensers have remained perfectly tight, and no boiler or superheater troubles have been experienced. The ordinary watch-keeping duties, such as regulation of feed-water, have involved no special effort on the part of the engine-room staff, and this has been confirmed by indirect inquiries made to elicit the general opinion of the junior personnel. In fact, there are no features arising out of the use of high-pressure superheated steam in association with the water-tube boilers which have occasioned the senior or junior engineers any concern. The temperature of the boiler-rooms and machinery spaces leaves nothing to be desired. Three or four main steam-pipe joints have shown slight leaks; these were remade, and no further trouble experienced. For success in this respect, ample elasticity should undoubtedly be provided, preferably in the form of "Aiton" corrugated bends, so as to relieve the flanges of undue stress. It is also desirable to avoid an accumulation of water in the pipe-lines, as this promotes temperature differences which are liable to stretch the bolts. With the turbo-generator and turbo-feed pumps taking their supply from the main lines, this is not likely to occur even under "standby" conditions. The fuel consumption registered on trial has been improved upon, due to setting the high-pressure turbines at their net designed clearances, and also attention to detail in various directions. The consumption of fresh water for make-up feed is 35 tons per day—about half the quantity used in earlier vessels of similar power. During the official trials, arduous tests were made of the maneuvering capabilities of the vessels, and the results on service in this respect have been admirable, sufficient steam being always available on demand.

The combination of oil-fired water-tube boilers and single-reduction geared turbines is an extremely flexible one, enabling a good spirt of speed to be maintained under favorable weather conditions, thereby overtaking detention due to adverse weather or fog. Most of the boilers have been opened up after two voyages, but some of them have been steamed for longer periods. A light-brown dust deposit, amounting to 20 ounces per boiler, distributed over the interior surfaces, has been removed, revealing the original graphite coating unimpaired. This deposit is ferric oxide, and is probably originating in the steel feed-pipes or boiler tubes, or both. There is room for some improvement in the de-aeration of the feed-water, and other corrective measures are being applied with the object of reducing or eliminating the oxide dust. There is a fair expectation that it will be possible to steam these boilers for six months without brushing the internal surfaces. The total expenses for cleaning the water side of the boilers have averaged about 10*l.* per voyage, as compared with 25*l.* in vessels (of 12,000 shaft horse-power) based on the same port, having 10 single-ended Scotch boilers, five of which are cleaned every voyage. Altogether the installations give every

indication of a high standard of reliability being attained on extremely low maintenance charges. In view of the general conclusions reached by Sir J. Biles in his 1925 paper, reference may, perhaps, be suitably made here to the economics of high-pressure steamships and motorships of this character. At the time the contract for the *Duchess of Bedford* was placed, prices were taken from leading shipbuilders for quadruple-screw Diesel installations, each set comprising eight-cylinder single-acting Sulzer-type engines. The quotations showed that the cost of a motorship of similar capacity would have been 100,000*l.* more. On the basis of 15 per cent for interest, depreciation and insurance, this represents an annual charge of 15,000*l.* Now, in comparisons which have hitherto been made there has always been a good deal of difference of opinion as to the relative cost of boiler and Diesel oils; the cost of repairs is also an awkward factor in such calculations. The writer proposes a close examination (Tables XI, XII and XIII). In the North Atlantic service, as oil is cheaper at American and Canadian ports than at United Kingdom ports, full bunkers should be taken at the former. In the Pacific service, similar remarks apply to Vancouver and Hong-Kong. While it is not permissible to quote actual contract prices, the prevailing market rates (Table XII), are quite appropriate for the purpose of comparison. Much has been said of the advantage of the smaller weight of bunkers required by the motorship; it is therefore interesting to examine that aspect of the question. A true comparison should include the combined weight of the propelling machinery, fuel, and feed-water as in Table XIII.

Having dealt with first cost, fuel costs, and weight of bunkers, the only other question of moment is repair expenditure (running and periodical). The sum of 200*l.* per voyage amply covers the running repairs in the *Duchess of Bedford's* installation.

	£
For eleven voyages per annum	2,200
Annual overhaul—B.O.T. survey	2,800
	£5,000

Examining the major charges, which conceivably may have to be faced in an installation of this kind (allowing the vessel a life of 25 years), and taking an ultra-cautious view of possible replacements:—

	£
Renewal of condenser tubes every seven years	7,000
Renewal of superheater tubes every seven years	3,600
Renewal of air-heater tubes every seven years	2,400
Renewal of boiler fire row tubes every seven years	3,300
	16,300
	3
	48,900
	Say, £50,000 in 25 years.

Let us say 2,000*l.* per annum.
Total repairs: 5,000*l.* plus 2,000*l.* = 7,000*l.* per annum.

The significance of these figures is that the annual charge on the extra first cost of the motorship, viz., 15,000*l.*, is sufficient to defray the entire cost of repairs (running and periodical) in the steam installation, with 8,000*l.* to spare. As the fuel costs of the steam plant in the two services quoted are not greater than the Diesel, it is not even necessary to inquire what the

TABLE XIII.

	18,000 S.H.P. Installation.			30,000 S.H.P. Installation.		
	Steam.		Diesel.	Steam.		Diesel.
Machinery complete	—	Tons. 2,275	Tons. 3,300	—	Tons. 3,410	Tons. 5,300
Fuel (plus two days' reserve) ..	(6,000 miles).. ..	1,190	1,465	(14,000 miles) ..	5,730	4,570
Feed-water	Half passage ..	250	Nil	Part passage ..	500	Nil
		4,435	4,765		9,640	9,870

TABLE XIV.

—	Speed, knots.	S.H.P.	Daily Consumption A.P. Converted to 18,500 B.Th.U's.	Daily Consumption Tons. Actual A.P.	$D^{\frac{2}{3}} V^3 / S.H.P.$	$D^{\frac{2}{3}} V^3 /$ Tons/day A.P.	Calorific Value of Coal.	Remarks.
							B.Th.U.s.	
Beaverburn, W.	15·52	6,600	(76·8)	87·0	319	25,000	11,900	Mech. stokers.
Beaverford, W.	15·10	6,500	(77·0) 76·9	81·2	297	23,900	12,800	Mech. stokers.
Beaverbrae, W.	14·90	6,560	(81·4)	80·2	283	23,000	12,700	Hand fired
Beaverhill, W.	14·50	6,000	(78·1) 79·7	80·0	278	20,500	13,200	Hand fired.
Beaverburn, E.	15·0	6,400	(74·7)	77·0	322	27,000.	13,100	Mech. stoker.
Beaverford, E.	15·2	6,370	(76·0) 75·0	84·8	330	24,300	13,100	Mech. stoker
Beaverbrae, E.	14·9	6,480	(81·3)	80·1	298	24,200	13,700	Hand fired.
Beaverhill, E.	14·8	6,188	(83·4) 82·0	84·0	320	23,600	13,620	Hand fired.

cost of repairs is in the latter. It may remain as a surplus of indeterminate magnitude. As it has been suggested, however, that there are some ports where the difference between the price of Diesel and boiler oil is small—say as little as 3*s.* per ton—it may be of value to examine the 18,000 shaft horse-power case on the basis that the difference is nil. The nearest approach to this appears to be San Francisco, where the market rates for Diesel and boiler oil are 33*s.* and 30*s.*, respectively.

	£	s.	d.	
Steam: 121 tons per day at 30s.	= 182	0	0	
Diesel: 92 tons per day at 30s.	= 138	0	0	
Difference in favor of Diesel	£44	0	0	per day.

On 200 days' steaming per annum, this equals 8,800*l.* The position thus becomes:—

	£	s.	d.	
Extra cost of fuel for steam plant	= 8,800	0	0	per annum.
Total repairs	= 7,000	0	0	"
	= 15,800	0	0	"
Charges for extra cost of motor vessel	= 15,000	0	0	"
	£800	0	0	

As the cost of lubricating oil for a Diesel installation is certainly in excess of this figure, the cost of repairs may again be left as an unknown surplus.

Some interest attaches to the performance of the five cargo steamships of the *Beaver* class, as being representative of one aspect of modern coal-burning practice. Commissioned a year ago, they are all twin-screw vessels of about 7000 horsepower, with Parsons' turbines, working pressure 250 pounds, temperature 650 degrees F., a proportion of the auxiliary machinery using steam superheated to a temperature of 550 degrees F. Three of the vessels have four Yarrow and two single-ended Scotch boilers, the Yarrow boilers being fitted with mechanical stokers of the "Erith-Roe" type. The other two vessels have four Babcock and two Scotch boilers, all hand fired. In each case, one Scotch boiler only is in use at sea. The results given in Table XIV have been registered under comparable weather conditions.

These installations were designed to give consumption rates of 1.1 pounds and 1.15 pounds of coal per S.H.P.-hour (all purposes) in the stoker and hand-fired jobs respectively, on a fuel calorific value of 13,500 B.T.U.'s. Both rates were comfortably realized on the official trials, and have been maintained on service. On trial, boiler efficiencies of 82 per cent and 75 per cent were obtained with the stoker and hand firing, respectively. On service, coal varying in calorific value between 11,000 and 13,000 B.T.U.'s has been burned on the "Erith" stokers, whereas good-quality bunker coal of 13,500 to 14,000 B.T.U.'s has been used in the hand-fired jobs. It will be observed from a study of the foregoing table that when the consumptions are correct to a uniformed basis of 13,500 B.T.U.'s the mechanically fired installations show a higher average efficiency of 9 per cent. The difference has been as low as 5 per cent and as high as 10 per cent on individual voyages. The vessels with the mechanical stokers have registered about ¼ knot to ½ knot higher average speed. The coal used in the stoker installations is a mixture of rough smalls and nuts. Purchased on a basis of calorific value, an economy of several per cent has been secured, as compared with the ordinary bunker coal used by the other vessels. The mechanical stokers, which are

supplied with air preheated to 310 degrees F., have given satisfaction. The ships leave one terminal port with the stokers and main engine maneuvering valves set, and make the Atlantic passage without any adjustment and with entire absence of variation in steam pressure. There is no noise, only a trace of smoke, and the working conditions are very easy for the firemen, whose duty consists of shoveling the coal into the hoppers. At a later date it is intended to fit suitable trunking to the cross and 'tween-deck bunkers so that the coal may feed to the hoppers by gravity. The total maintenance charges are less than 1*d.* per ton of coal burned. These expenses have been incurred in renewals of cast-iron grate parts. In certain of the stokers the grate parts were made of Hadfield's "Era" heat-resisting steel, and have been in use for twelve months without deterioration. Experience has shown that it is possible to burn coals having ash of a low fusing temperature on these grates without damage. The renewals to brickwork have averaged about 5*l.* per voyage.

The coal as fed to the stokers has to be less than 2-inch cube, and supplies are arranged accordingly. In services where this can be conveniently done, the use of mechanical stokers is advantageous. The charges on their first cost amount to 5*d.* per ton of coal burned plus 1*d.* for maintenance, representing, with coal at 20*s.* per ton, a deduction of 2½ per cent from the gross saving previously indicated. Table XV giving a comparison between the *Beaverburn* and *Minnedosa* illustrates the fuel rates of two relatively efficient types of steam plant, burning coal.

The *Beaverburn* cannot, of course, be regarded as a high-pressure instalation. With a working pressure of 350 pounds and steam temperature of 700 degrees F., the fuel rate would be 0.94 pound per shaft horsepower per hour. With oil of 19,000 B.T.U.'s calorific value, and a boiler efficiency of 85 per cent, this figure would be about 0.63 pound for propulsion. Now a large number of modern cargo and intermediate vessels fall into two classes, viz.:—Single-screw ships, 4000 to 6000 shaft horsepower. Twin-screw ships, 7000 to 10,000 shaft horsepower. The fuel rate of 0.63 pound shaft horsepower per hour for "Propulsion," which does not, in this class of vessel, differ sensibly from the "All purposes" figure, may be taken as applicable to both types. Consider this in relation to a Diesel installation. Putting aside for the moment the question of burning coal in the steamships (Tables XVI and XVII).

Example.—Twin-screw cargo vessel, 7000 shaft horsepower, 15 knots—bunkering radius, 10,000 miles.

	£	s.	d.
Cost of motorship	300,000	0	0
Cost of steamship	270,000	0	0
Difference	£30,000	0	0

15 per cent (interest and depreciation) on difference = 4,500*l.*

Average prices of boiler and Diesel oil, say 50*s.* and 60*s.* per ton respectively.

Then daily fuel costs at 7000 shaft horsepower:—

	£	s.	d.	
Steamship = 47.5 × 50s.	= 118	15	0	per day.
Motorship = 31.5 × 60s.	= 94	10	0	"
Difference in favor of Diesel	£24	5	0	"

On 200 steaming days per annum this equals 4,850*l.*

TABLE XV.

—	Type of Machinery.	Boilers.	Thermal Conditions at Engines.			Rate per S.H.P. hour (Propulsion).	Calorific Value of Fuel.	Boiler Efficiency in Service.
			Pressure.	Temp.	Vacuum.			
				Deg.	In.			Per cent.
Beaverburn .. (S.H.P.=6,600)	S.R. turbines. Steam auxiliaries	Water-tube with mechanical stokers and Scotch boilers, hand fired	235	630	29	1·03	13,500	80
Minnedosa .. (S.H.P.=11,500)	Reciprocating engines and L.P. turbine (direct drive). Steam auxiliaries	Scotch boilers, hand fired	208	530	29	1·08	13,500	74

TABLE XVI.

—	Oil Fuel per S.H.P./hour (Propulsion).
High-pressure steam—4,000 to 10,000 shaft horse-power	0·63
Diesel—4,000 to 10,000 shaft horse-power ..	0·42

TABLE XVII.

—	Steamship. (Oil-burning, 2 Water-tube and 1 Scotch Boiler. Diesel and Turbo-generators.)	Motorship.
	Tons.	Tons.
Machinery complete	1,000	1,400
Fuel for 10,000 miles (plus 2 days' reserve)	1,405	939
Feed-water (part passage) ..	200	Nil
	2,605	2,339

If the economics of high-pressure steamships and motorships of this class be fully explored, it will be found that in the majority of cases the advantage lies with the former, this condition holding when the steamship is using oil as fuel, the extra cost of boiler oil per annum only being slightly in excess of the annual charge on the higher cost of the motorship. It is evident, therefore, that if the steamship is also able to use coal in zones where its price is attractive, a strong superiority obtains with that type of installation. It would therefore remain for the naval architect to design such ships so that a deep tank might be utilized for coal or cargo, and double-bottom tanks for oil or ballast, and for the engineer to initiate suitable stokehold arrangements, enabling pulverized coal or oil to be used alternatively and in a manner compatible with safety. The difficulties of this may be considerable; nevertheless it is thought that they are susceptible of solution. The future of steam propulsion is bound up with further advances in thermal efficiency and reductions in the first cost of the steamship. On the basis of our experience to date there does not seem to be reason to anticipate any practical difficulty with pressures of 500 pounds to 600 pounds, and temperatures of 750 degrees F., but progress towards these conditions would naturally come first in smaller ships. If power station practice is stabilized at or about 1000 pounds pressure and temperatures of 800 degrees F. to 900 degress F., it is scarcely conceivable that marine practice will remain at 220 pounds and 600 degrees F. Apart from a rise in pressures and temperatures, the next step of real commercial value appears to be a radical advance in water-tube boiler design, resulting in substantial economies in weight and cost, in association with efficiencies approaching 90 per cent based on the gross calorific value of the fuel. It is known that such developments are being prosecuted in this country and elsewhere, and these, if successful, will enable large capacities per unit to be easily obtained, resulting in a general simplification of the steam-raising apparatus. The battery of boilers and their appurtenances has hitherto been the chief disability in steam plant, everyone recognizing the manifest advantages of turbines and other rotary mechanism. The water-tube boiler fully developed for marine work will enable injected fuel, solid or liquid, to be employed at option. On the basis of experience to date with high-pressure plant, such development work as the foregoing may be expected to yield consumption rates of the order of 0.5 pound of oil for "Propulsion" and 0.55 pound for "All purposes" in high-power installations. If the most is to be made of the inherent merits of steam propulsion, the sum total of the advantages must be grasped, as expressed in low first cost, minimum weight of machinery and bunkers, minimum fuel cost (with choice of pulverized coal or oil), and, finally, low maintenance charges. There appears to be a movement towards higher speeds both in cargo and passenger services, and this will, in many instances, be accompanied by the use of larger vessels. The power of the propelling machinery will advance commensurately, and, all circumstances considered, high-pressure steam propulsion may be expected to occupy a predominant position.

APPENDIX.

Section 1.—As usage varies somewhat, it is necessary to define the precise meaning of the coefficients employed in the paper. Under the heading "Propulsion" is included all fuel expended in working the main engines and the whole of the auxiliary machinery (including steering gear) incidental to the propulsion of the vessel. "All Purposes" referred to the *total* weight of fuel consumed per day, and embraces that defined for "Propulsion" plus that expended in lighting, heating, cooking, refrigeration, ventilation, etc., or

what may be appropriately described in ships of this class, as "Hotel Services." The Admiralty and Fuel Coefficients are used for comparative purposes, within permissible limits only, for ships of somewhat similar proportions and speed subject to North Atlantic weather conditions. These coefficients are in general use nowadays for interpreting the service performances of ships. They are used to supplement the "fuel per shaft horsepower-hour" ratio because the cost of producing power cannot properly be considered apart from the efficiency of propulsion. If the prime mover is of a heavy and costly type, the natural tendency is to put up the revolutions, resulting in a loss of propeller efficiency. Comparisons between power station and marine practice are both suitable and desirable up to a point, but the overall performance of a ship must ultimately be assessed by appropriate technical and commercial standards.

Section 2.—As frequent reference is made to the steam rates of turbines, boiler efficiencies, etc., it is desirable to explain the methods adopted during trials to obtain accurate data. Two torsion-meters of the Siemens-Ford type were fitted to each line of calibrated shafting and the zeros carefully set. The engines were then worked up the power scale, and the meter readings noted. Close correspondence was obtained and the small differences reduced to a minimum by averaging the results. A Glenfield and Kennedy meter of the displacement type was fitted on the discharge side of the feed pumps. A test was run for some hours with only the main turbine condensate passing, so as to register the net steam rate. Certain deep oil tanks were reserved for the fuel consumption trials, and these were sounded and Pneumercator readings taken at an anchorage in calm water before the start of the trials, and again at the completion, after the vessel returned to an anchorage. Careful Pneumercator readings of fuel consumption were taken hourly during the trials and the boiler efficiency calculated from the meter readings and other boiler data, a particularly full set of which was recorded. It should be observed that the boiler efficiency of 85.5 per cent was registered in association with a steam rate of 7.3 pounds for the turbines. Obviously, if the latter figure were too good, the boiler efficiency would be correspondingly augmented, and conversely, if the boiler efficiency were too high, the steam rate would be correspondingly diminished. The fact is that these figures check one another. In a similar manner, the shaft horsepower as used in the machinery calculations was applied to the propulsive coefficients of the vessel. All data relative to the performance of the various units in the installation were cross-checked with the utmost care.

Section 3.—In view of the fact that important deductions are based on the relative prices of a Scotch and water-tube boiler of similar capacity, the weights of the two types are shown in greater detail in Table XVIII, in order that the features of each may be properly focused. Opinion differs as to the durability of Scotch and water-tube boilers. This is a matter which can best be dealt with by discussion, but comparison should, of course, be made between up-to-date examples of each.

It may be observed, however, that the water-tube boiler is one having a definite mode of circulation with no appreciable temperature difference existing in its component pressure parts. It has no stays, and in the high-pressure type there are no rivets or caulked landings. The pressure parts, including the tubes, are of stout scantlings, and if made of good material and provided with suitable feed, possess all the elements of great durability. With good design and a suitable system of forced draught, there are no grounds for apprehending trouble or undue deterioration of the casings. With injected fuel, the risk of major damage to the boiler tubes is small, as the fuel can be shut off immediately if the water is lost. Under the

TABLE XVIII.—Comparison of Scotch and Water-Tube Boilers of Similar Capacity.

—	Empress of Australia. (Double-ended S.B. 220-lb. Working Pressure. 620 deg. F. Temperature. Heating Surface, 6,250 sq. ft. Oil Basis.)	Water-tube-Boiler. (350-lb. Working Pressure. 680 deg. F. Temperature. Heating Surface, 5,200 sq. ft.)
	Tons	Tons
Total weight ex water	130	73
Weight of water	49½	7
Working weight	179½	80
Steel generator pressure parts	95·00	24·50
Superheater	3·00	5·60
Air heater	10·00	13·50
Light steel-work	Smoke-boxes and up-takes to base of funnel.	All boiler casings and uptakes to base of funnel
	10·00	12·80
Brickwork	1·50	13·10
Mountings	2·50	1·00
Lagging	4·00	2·50
Furnace fronts	4·00	Included in casings
	130·00	73·00

stimulus of modern requirements, the design of automatic feed-water regulators is undergoing changes in the direction of closer control and greater reliability, all conducive to ease and safety of operation. The heat transmission rates in mercantile water-tube boilers are low by comparison with naval and power-station practice, and there is no reason why the life of the fire rows of tubes should not extend to ten years or longer. The brushing of the interior of the tubes, where scale is absent (as it ought to be) is an extremely simple process and can be effected expeditiously and cheaply. Contrary to the views one often hears expressed, the Scotch boiler is not easy to clean thoroughly, particularly in way of tube necks, tube plates, and combustion chamber back plates, where space is restricted. The large double-ended Scotch boiler, as constructed for pressures of 220 pounds, is a credit to the boiler-maker, but the fact remains that it is a primitive device for raising steam, and when the multitude of stays, large and small, flat surfaces, collapsing pressures on furnaces, deformation of combustion chambers, and other characteristics, are considered in their entirety, it cannot be seriously considered for higher pressures alongside the modern water-tube boiler, particularly for large capacities, which, unquestionably, is the direction in which development work must proceed if the economic possibilities inherent in steam plant are to be utilized.

Section 4.—The forced-draught arrangements in the *Duchess of Bedford* and later installations are worthy of notice. Each boiler has its own motor-driven forced-draught fan (of very ample capacity), situated alongside the boiler at the stokehold platform level. The trunking arrangement is therefore simple and difficulties of distribution between boilers eliminated. The total volume of air required for combustion traverses the stokehold platforms, resulting in very comfortable working conditions. As each boiler is a separate unit with its own uptakes and air heater, it may be opened up, if necessary, for attention or cleaning on the fire side at sea. In the later and larger installations, where the boilers are of 6000 H.P. and 8000 H.P. capacity, each boiler has its own forced-draught and induced-draught fan, the latter being motor driven and placed above the air heater. These boilers will be worked under balanced draught. Whatever merits the closed stokehold system may have for naval work, it lacks many of the features which the foregoing arrangement provides for mercantile purposes. With balanced draught there is no tendency for stokehold air to leak into the furnace, or for the furnace gases to leak into the boiler-rooms, the ventilation of which may be supplemented according to the nature of the climatic conditions under which the vessel is to be operated.

Section 5.—Although great advances have been made in condenser tubes, marine engineers will naturally welcome further security. In view of the combined efforts now being made by manufacturers and designers, there is reason to believe that material progress in this direction will be made. In the high-pressure installations under review, reserve distilled water-tanks are fitted in the wings of the engine room at a level above that of the condensers. Their capacity is in excess of the internal volume of the condensers, and pipe connections are provided to enable the contents to be discharged into the condensers and pumped back again after the water test is completed. This operation occupies 30 minutes. There are also connections to the main hot-well tank with control valves at the starting platform, so that the combined capacity of the reserve tanks is always readily available in emergency. The arrangement is illustrated on Fig. 3.

Section 6.—The quantity of steam piping in these high-pressure steam installations is surprisingly small. The adoption of electrically-driven auxiliaries, of course, eliminates a great deal in the engine room. The maze of

piping one is accustomed to see on the tops of a battery of Scotch boilers is absent. The total weights of steam and exhaust piping in the *Empress of Canada* and *Empress of Japan* boiler rooms will be 9 tons and 6 tons, respectively. With large capacity boilers, the number of pipes, valves, joints, and fittings is sensibly diminished. In vessels with two or more boiler rooms the major proportion of the boiler power may be arranged in the aft compartment, forward and adjacent to the engine-room.

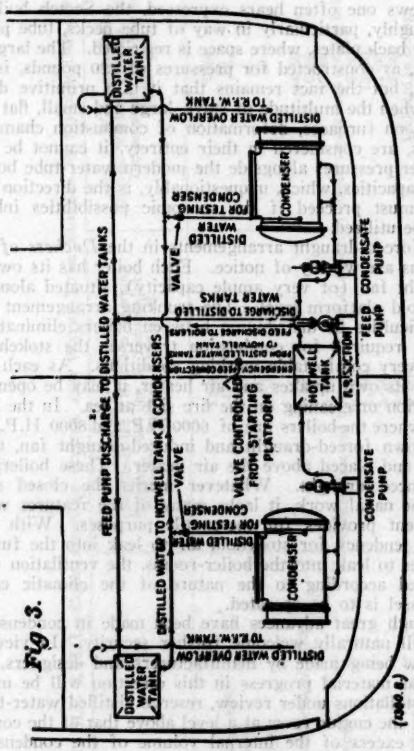


Fig. 3.

DIAGRAMMATIC ARRANGEMENT OF CONDENSATE AND FEED PUMPS SUCTIONS AND DISCHARGES, SHOWING EMERGENCY MAKE-UP FEED FROM DISTILLED WATER TANKS TO HOT-WELL TANKS AND CONNECTIONS FOR TESTING CONDENSERS WITH DISTILLED WATER.

Section 7.—It is not sufficiently appreciated in certain quarters how much stress is laid nowadays on the elimination of vibration. What was accepted some years ago would not be tolerated today. New standards have arisen, and the requirements of the traveling public become more exacting daily. An examination of the claims of various types of propelling plant is not complete without considering the effect on seatings and the hull structure throughout the life of the vessel, and the possible expense involved in rectifying defects.

Section 8.—The combination of Diesel and turbo-generators referred to in the text has many attractions. The use of electrically-driven auxiliaries throughout the machinery spaces and the adoption of electricity for cooking, added to the growing requirements in the matter of lighting, heating and ventilation, constitute in the aggregate a very substantial figure. The auxiliary Diesel generator is admirably suited for this duty, and also permits of boilers being shut down entirely in port in all excepting extreme winter conditions. As a vessel may be in port 160 days out of the year, a considerable saving in fuel is effected. The Diesel generators are, however, very costly, and it has been found expedient to limit their capacity to the average sea-load. The cost of providing adequate reserve with this form of plant would be almost prohibitive, but having regard to the paramount importance of electrical energy being available at all times for propelling auxiliaries, steering gear, etc., adequate reserve there must be. This can be most cheaply and conveniently provided in the form of turbo-generators arranged to work with high-pressure superheated steam. These machines are relatively light, and take up a modest amount of space, so that 100 per cent standby (on the average load) can be suitably provided. At sea the Diesel generators may take two-thirds of the load and the steam sets one-third. Under maneuvering conditions, or in narrow waters, both types of machine are always run in order to minimize the risk of failure of current to any of the important auxiliaries. The switchboard is designed so that essential auxiliaries always receive current in preference to the domestic services in the event of there being a shortage, due to the stoppage of a machine. The use of turbo-generators in association with the turbo feed-pumps during maneuvering is helpful in promoting a circulation of steam through the superheaters and steam pipes.

Section 9.—The use of coal and oil for marine purposes has recently been dealt with by Sir Eustace Tennyson d'Eyncourt and others, and very little remains to be said on the subject. As a matter of general interest, the following tables have been compiled to show the amount and distribution of the known coal and petroleum resources. It is not claimed, of course, that these are anything more than indicative in a very broad way of the great disparity between the world's coal and oil reserves.

COAL.	Millions of Tons.
Great Britain	147,000
United States	3,836,657
Germany	410,000
France	19,000
Belgium	20,000
Austria-Hungary	17,000
Roumania	40,000
Australia	344,000
Japan	48,000
China	630,000
Canada	1,234,269
South Africa	56,200
India	.....
Total	6,802,126

U.K. PORTS
B.O. 67/6
D.O. 82/6
COAL 18/-
VANCOUVER
B.O. 39/
D.O. 52/
COAL 27/
SAN FRANCISCO
B.O. 38/
D.O. 54/6
COAL 33/
MONTREAL
B.O. 39/6
D.O. 53/6
COAL 30/
NEW YORK
B.O. 28/6
D.O. 57/6
COAL 28/-
KINGSTON
B.O. 34/
D.O. 52/-
COAL 31/6
BUENOS AIRES
B.O. 60/-
D.O. 75/-
COAL 42/6
VALPARAISO
B.O. 52/
D.O. 73/-
CAPETOWN
B.O. 82/6
D.O. 95/-
COAL 34/6
DURBAN
B.O. 80/-
D.O. 95/6
COAL 29/6
SUEZ
B.O. 72/6
D.O. 87/6
COAL 32/6
ADEN
B.O. 72/6
D.O. 77/6
COAL 30/6
COLOMBO
B.O. 75/-
D.O. 90/-
COAL 36/-
BOMBAY
B.O. 75/-
D.O. 80/-
COAL 29/6
CALCUTTA
B.O. 75/-
D.O. 90/-
COAL 30/-
SINGAPORE
B.O. 70/-
D.O. 75/-
COAL 33/
HONG KONG
B.O. 70/-
D.O. 75/-
COAL 36/6
JAPAN
B.O. 70/-
D.O. 75/-
COAL 30/6
MELBOURNE
B.O. 77/6
D.O. 82/6
COAL 41/3
SYDNEY
B.O. 77/6
D.O. 82/6
COAL 33/6
BUNKERING PORTS
B.O.-BOILER OIL
D.O.-DIESEL OIL
(5000 c)

Fig. 4.

PETROLEUM.	Millions of Barrels.
United States	7,000
Canada	995
Mexico	4,525
N. and S. Africa	9,280
Algeria and Egypt	925
Persia and Mesopotamia	5,820
S. E. Russia	5,830
Roumania and Galicia	1,135
N. Russia and Saghalien	925
Japan and Formosa	1,235
China	1,375
India	995
E. Indies	3,015
Total	43,055=
	6,900 million tons.

On the basis of the foregoing comparison, there would appear to be something about one thousand times more coal by weight than oil. Moreover, according to the United States Geological Bureau, only 33 per cent of the petroleum is available as heavy fuel, the balance being disposed of in more lucrative markets.

Twin-Screw Cargo Vessel, 7,000 S.H.P. (15 knots).

	Diesel.	High-pressure Steam.	
		Boiler Oil.	Coal.
	Tons.	Tons.	Tons.
Daily consumption	31·6	47·5	71·0
Bunkers for 10,000 miles, plus 2 day's reserve—total 30 days ..	950	1,420	2,130
Machinery, bunkers and feed-water	2,350	2,620	3,330

There can be little doubt that for large high-speed passenger ships the only fuel to be entertained at present is oil. In the case of cargo ships considerations of cleanliness, speed of bunkering, etc., are not paramount, but the cost of fuel is.

The distribution of coal and oil and the corresponding prices at world ports are matters of considerable importance. An examination of the case of the cargo steamer referred to in the paper may profitably be extended, with particular reference to the burning of boiler oil or coal. On most of the trade routes, a steaming radius of 10,000 miles will span cheap zones of coal or oil. In order to illustrate the significance of this, the daily consumption for a vessel of this type is given in the above table, and the prevailing market rates at the principal ports are shown in Fig. 4.

Using these consumption rates, one may readily compute the relative fuel costs on any of the well-known services. To be free to utilize coal or oil, whichever is cheapest, at any stage of a voyage will, it is submitted, enable the voyage to be made on the lowest possible expenditure for fuel.—"Engineering," April 5, 1929.

H. M. BATTLESHIPS *NELSON* AND *RODNEY*.*

By Sir William Berry, K.C.B.

The terms of the Washington Treaty permitted this country to proceed at once, after its ratification, with the building of two capital ships, and with this exception, all capital ship construction by the Powers signatory to the Treaty was suspended until 1931. It is therefore hoped that an account of these vessels will be of interest to this Institution, and will provide a fitting sequel to the papers read in 1919 and 1920, by former Directors of Naval Construction, namely, "Ships of the British Navy on August 4, 1914,"† by the late Sir Philip Watts, and "Naval Construction during the War,"‡ and "H.M.S. *Hood*,"§ both by Sir Eustace Tennyson d'Eyncourt. Before proceeding with the *Nelson* design, it may be of interest to place on record a short summary of the history of capital ship design and construction for the British Navy between the time of building H.M.S. *Hood* and the laying down of H.M.S.S. *Nelson* and *Rodney*. During this period, a large number of designs were prepared for the consideration of the Board of the Admiralty, embodying the lessons of war experience and the recommendations of the Post-War Questions Committee, as a result of which orders were placed by the Board in October, 1921, for four battle cruisers, the shipbuilding firms selected being Messrs. John Brown, of Clydebank; Messrs. Fairfield, of Govan; Messrs. Swan, Hunter, of Wallsend, and Messrs. Beardmore, of Dalmuir. These ships had an armament of nine 16-inch guns in three triple turrets and sixteen 6-inch guns in eight twin turrets, and a speed of 32 knots, requiring a shaft horsepower of 160,000. The legend displacement on the basis then in use was 48,000 tons, the corresponding standard displacement as defined by the Washington Treaty being 47,540 tons. Their size was limited by the dimensions permitting passage through the Panama and Suez Canals, and by the docking facilities at Rosyth and Portsmouth Dockyards. In succession to these battle cruisers there was under consideration a battleship design, with a displacement about 500 tons greater, carrying nine 18-inch guns and sixteen 6-inch guns, with a speed of 23½ knots. Shortly, however, after the orders for these four battle cruisers had been placed, and whilst the design of the more powerful vessel referred to above was still under consideration, the negotiations which culminated in the signing of the Washington Treaty took place. The construction of the battle cruisers was suspended in November, 1921, before any work had actually begun, and the building contracts were definitely cancelled in February, 1922.

As affecting capital ships, the relevant conditions imposed by the Treaty are:—(i) That the standard displacement of a capital ship shall not exceed 35,000 tons; (ii) "No capital ship * * * shall carry a gun with a caliber in excess of 16 inches." The influence of these conditions on design was far-reaching. For the first time it was necessary to work to a fixed and absolute limit for displacement which on no account might be exceeded, although it was almost equally important to approach this limit very closely in order to achieve the maximum possible results. It was, therefore, all-important to seek out and adopt every possible means of saving weight, and, moreover, to estimate the weight with a very high degree of care and accuracy.

Article II of the Treaty allowed Great Britain to proceed at once with the construction of two new capital ships. The design for the vessels was prepared and the necessary drawings and specifications made during 1922, Sir

* Paper read before the Institution of Naval Architects, London, on Wednesday, March 20, 1929. Abridged.

† See "Engineering," vol. cvii, page 479. (1919.)

‡ See "Engineering," vol. cvii, page 482. (1919.)

§ See "Engineering," vol. cix, page 423. (1920.)

Eustace d'Eyncourt being Director of Naval Construction. The principal particulars of the design, as finally approved, are as follows:—Length between perpendiculars, 660 feet; length overall, 710 feet; breadth, extreme, 106 feet; draught in standard condition, 30 feet; displacement (standard), 35,000 tons; freeboard, forward, 29 feet; freeboard, amidships, 25 feet 6 inches; freeboard, aft, 27 feet; shaft horsepower, 45,000 (2 shafts); speed, 23 knots; revolutions, 160 per minute; oil fuel capacity, 4000 tons; complement (as flagship), 1400. The approved armament is as follows:—Nine 16-inch guns in three turrets; twelve 6-inch guns in six turrets; six 4.7-inch anti-aircraft guns; eight 2-pounder pompoms; two submerged torpedo tubes. The armor consists of, main belt, 14 inches; barbettes, 15 inches; turrets, 16 inches maximum; deck, 6¼ inches.

Orders for the ships were placed, on December 11, 1922, with Messrs. Sir W. G. Armstrong, Whitworth and Company for *Nelson*, and with Messrs. Cammell Laird and Company for *Rodney*, the keel plates being laid for both ships on December 28, 1922. The machinery for the former vessel was supplied by Messrs. The Wallsend Slipway and Engineering Company, and for the latter by the shipbuilders. Messrs. Armstrong, Whitworth supplied the 16-inch gun mountings for both vessels, and the 6-inch gun mountings for *Nelson*. Messrs. Vickers Limited, supplied the 6-inch gun mountings for *Rodney*. H.M.S. *Nelson* was launched on September 3, 1925, and was completed on September 10, 1927. The corresponding dates for H.M.S. *Rodney* were December 17, 1925, and November 10, 1927.

The limiting gun caliber of 16 inches imposed by the Treaty brought into the British Navy a weapon of a size not hitherto in use. New designs of guns and mountings, therefore, had to be prepared. The economy in weight to be obtained by mounting the guns in triple instead of twin gun turrets had been appreciated and utilized in the intermediate designs previously described, and in the design to which *Nelson* and *Rodney* were eventually built the triple turret was retained. Three of these turrets could be carried on the standard displacement, and the grouping of these three turrets forward (the center turret being superimposed), which had been accepted for the abandoned battle cruisers, was also followed in the *Nelson* and *Rodney*. With this grouping of the main armament is associated a minimum length of armored citadel, so allowing the maximum thickness of armor to be fitted, whilst efficient protection to the main magazines could also be provided. The advantages so gained were considered to outweigh the loss of fire directly astern necessitated by the arrangement, which, in conjunction with the consequent displacement of the bridge structure to a positon abaft mid-length, gives these vessels their very distinctive appearance. In order to avoid smoke interference with the controls, the boiler-rooms were placed abaft the engine-rooms, so enabling the funnel to be abaft the bridge structure.

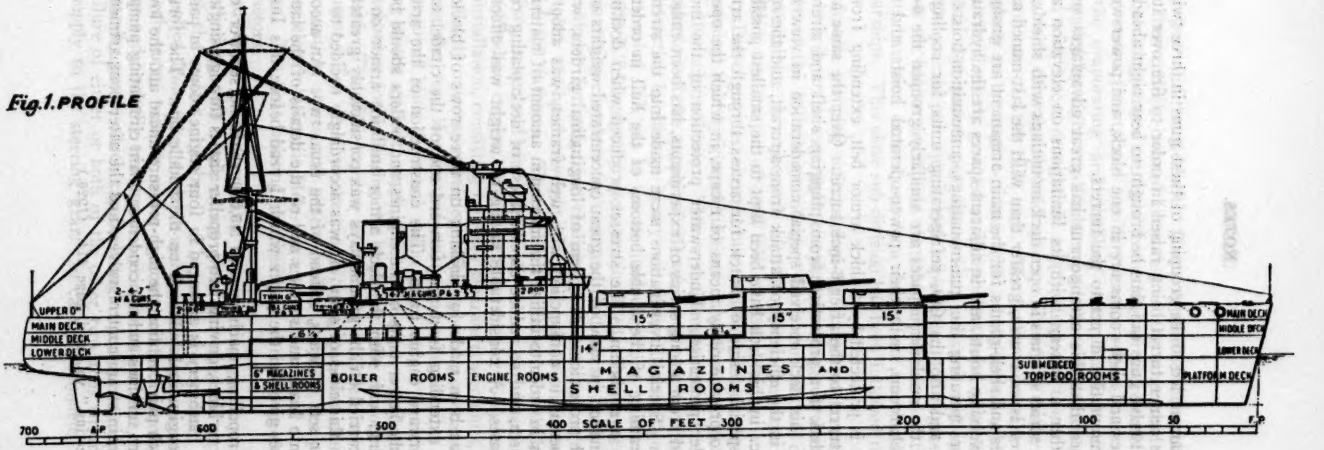
The problem of weight reduction was attacked in two directions—firstly, the general design of the ship was so disposed as to give the maximum effectiveness to armor and armament within the smallest possible compass; secondly, by close attention to the details of structure, machinery, and fittings, to eliminate all non-essential parts, and to use the lightest materials appropriate to the work. By careful consideration of all details (involving the re-design of many of the Admiralty standard pattern fittings), and by tightening up the supervision of manufacture, a large measure of success was achieved. One of the most fruitful sources of weight economy was found in the employment of "D" quality steel in place of the "H.T." quality formerly used by the British Admiralty for the principal strength members of the hulls of heavy warships. A description of the development of this

steel was given in a paper entitled "Steels for Shipbuilding," read by the writer before the Institution of Civil Engineers' Engineering Conference, 1928 (see "Engineering," vol. cxxv, page 720). This steel has an ultimate tensile strength of from 37 to 43 tons per square inch, with a minimum elastic limit of 17 tons per square inch; about 6500 tons of this steel was used in the hull structure of each ship, and by using it higher stresses were accepted, and consequent reduction of scantlings were made. The specified rolling limits were also amended to ensure that no plates or sections supplied should be above the weight ordered. Extensive use was made of aluminium and its alloys in the manufacture of minor fittings and of such articles as kit lockers, etc.

A number of different alloys were employed, some to proprietary formulæ and some of Admiralty specification, and experience so far has shown that in the main they have fulfilled their purpose. With further improvements in design and material, these alloys should have very many useful applications for fittings not exposed to the weather, nor subject to very rough usage. For outboard fittings, which are exposed to the corroding influences of sea air and spray, the results have so far been disappointing, and it is evident that further developments will be necessary before a really reliable light alloy for such work is obtained. The weather decks were sheathed with Douglas fir in lieu of the customary teak, the loss in wearing qualities and appearance being sacrificed for lightness. Plywood was largely used for dwarf and divisional bulkheads, not of importance for strength purposes. The heavier standard articles of furniture were re-designed and very considerably lightened, and here again plywood was extensively used. All wood fittings were fireproofed by the "Oxylene" process.

It will be seen from Fig. 1 that the ships have a flush weather deck. This gives greater freeboard aft, and increases the space available for accommodation as compared with the more usual arrangement of breaking down one deck abaft the mainmast. Rather more than half the length of the weather deck is swept by the main armament, the total arcs of training of the forward, middle and after turrets being respectively 298, 330 and 250 degrees. Special measures have been taken to protect the personnel and instruments on the bridges and control platforms from the blast of the after turret, which is very severe when the guns are trained aft at full elevation. The conning tower, with a revolving armored director position, is placed immediately abaft the main armament, and astern of this, again, rises the bridge structure. Fire-control systems have developed very greatly in recent years, and many difficulties have had to be overcome in providing the director towers with adequate support and the maximum possible clear view for their sights and rangefinders. The bridge and foremast have therefore coalesced into a very lofty and massive structure, bearing little resemblance to the arrangements previously adopted. This structure carries the directors for the 16-inch, 6-inch, and anti-aircraft guns, a bridge for the admiral and staff, and for torpedo control, a captain's bridge with signaling position and navigating platforms, and below these, various sea cabins and offices. The superstructure encircling the base of the bridge structure, the funnel and the main mast, accommodates the anti-aircraft and machine-guns, while the central space is occupied by the funnel, the boiler-room air intakes and various workshops. Four 36-inch searchlights are carried, one on either side of the funnel, and two on a platform on the mainmast. The boats are stowed amidships, between the mainmast and funnels, and are handled by a derrick operated by electro-hydraulic variable-speed gear winches. Placed above the weather deck, on the quarters, is the secondary armament of twelve 6-inch guns. Considerations similar to those advanced in the case of

Fig.1. **PROFILE**



H. M. BATTLESHIPS "NELSON" AND "RODNEY".

the main armament led to the grouping of these guns in three twin turrets on each side, the center turret being raised in order to fire over its neighbors. Eight of the twelve guns can thus be brought to bear right ahead and astern. The magazines and shell-rooms are in one block, and power-worked hoists supply the ammunition direct to the turrets.

It is claimed that this arrangement has great advantages over systems employing either a battery, with its limitations on elevation and arcs of training, or a series of guns in open deck mountings with shields, though the weight involved is, of course, greater than with the last-named arrangement. The magazines and shell-rooms for the main armament are grouped together around the revolving hoists, and in adjacent spaces are the hydraulic pumping machinery for operating the gun-mountings, ammunition-hoists and transporting gear, and also the CO_2 refrigerating units for cooling the magazines. Abaft the machinery spaces are similarly grouped the 6-inch magazines and shell-rooms, with their power-operated hoists and independent cooling plant.

The citadel is protected by a thick armor belt, extending from the foremost 16-inch turret to the after 6-inch turret. Over the same area extends an armored deck for protection from plunging shell and aircraft bombs; the design of this deck received special consideration in view of probable developments in the efficiency of attack from aircraft, and the openings for ventilation, etc., in this deck have been kept to the smallest possible size and number. A special design of hatch for access through the armored deck was evolved to provide ready means of escape, in which the operating gear is below, under protection. Under-water protection of the most efficient type, developed from a long series of experiments, has been embodied in the hull structure. Special investigations were made into the strength of the plating and framing of the double bottoms of the hull in order to reduce the weight to a minimum. The stresses produced when docking received careful examination in view of the great concentrated weights of the main armament and protection. A system of longitudinal girders, with widely-spaced deep beams in conjunction with web frames, was adopted for the principal decks in order to utilize the maximum amount of material for longitudinal stresses, and so reduce the thickness of deck plating required for strength purposes. A considerable saving of weight was effected by this means.

Provision has been made for docking on three rows of blocks, the side docking keels extending from the forward end of the citadel to the after end of the forward engine room. The construction of the armored deck required that the larger units of the engines and boilers should be on board before launching, and, with the deck armor and side armor on board, the weight to be carried on the groundways was considerably greater than had been usual for this class of ship. It was accordingly decided to launch the vessels on four sets of ways, instead of the usual two. An account of the investigation into loads and stresses and of the details of the launching arrangements was given in the paper which I read before this Institution in 1926.*

The main propelling machinery of 45,000 shaft horsepower consists of Brown-Curtiss turbines, driving two propeller shafts through single-reduction gearing. This machinery is arranged in four engine-rooms in pairs abreast, divided by a longitudinal bulkhead on the middle line. The forward compartment of each pair contains the high-pressure ahead and the low-pressure ahead and astern turbines and the condenser, main circulating pumps, starting platform and main steam-control valves, and the after compartment contains

* See "Engineering," vol. cxxi, page 414. (1926.)

the reduction gear and thrust-blocks. The complete set of machinery driving one shaft is thus entirely isolated from that of the other shaft. For use at low powers, cruising stages are fitted in the high-pressure turbines. Further aft in wing compartments are auxiliary engine-rooms, one on each side, containing the evaporating and distilling plant, auxiliary condenser, etc. Steam is supplied by eight small-tube type boilers fitted with superheaters and provided with forced-draught air supply, arranged two in each of four boiler-rooms, with central stokehold spaces. Six of the boilers are of 5800 H.P. each, and two of 5100 H.P. each. The boiler-rooms are arranged two abreast, similarly to the engine-rooms, with a longitudinal middle-line bulkhead, and can be similarly grouped as independent units on each side. Steam is generated at 250 pounds per square inch pressure, and supplied to the engines at 200 pounds per square inch superheated 150 degrees F. It will be seen from the above description that the sub-division of the machinery compartments has been carried out to a degree not previously attained in our large warships. This close sub-division has been followed throughout the ship.

Three hydraulic pumping units, driven by compound steam engines, are fitted, each in a separate watertight compartment under protection, for supplying power to the 16-inch mountings. Each unit has an output of 150 cubic feet per minute at a pressure of 1250 pounds per square inch. Six dynamos, with a total output of 1800 KW. at 220 volts, are provided. Two are driven by Diesel engines, situated one each side abreast the boiler-rooms, and the remaining four by geared turbines, in well-separated compartments forward of the engine-rooms. This arrangement ensures that the least possible proportion of the supply of electric power to the ring mains shall be put out of action by damage in any one spot. Electrically-driven refrigerating plant, using CO_2 gas as the medium, consisting of seven compressor units of 60,000 B.T.U. capacity each, are installed. Two units are employed for the refrigerating chambers for meat and food storage, and for icemaking, and the remainder for magazine cooling. The anchors carried are three No. Byers plan 175-hundredweight stockless pattern, with a chain cable equipment of 500 fathoms. The cables are made up with lugless joining shackles. Two bower and one sheet cable-holders are fitted. The former, with the middle-line capstan, are driven by an electro-hydraulic variable-speed gear installation, consisting of three separate power units, which can be used independently or cross-connected. This type of gear has the advantage of eliminating the very long run of steam and exhaust pipes which would have been required with the usual steam gear, and so avoiding the consequent heating of the spaces through which these pipes would pass. A special counting mechanism for indicating the length of cable run out at any instant has been fitted. The after capstan is driven by a Hele-Shaw type electro-hydraulic motor. Ten deck winches for working paravanes, embarking torpedoes, ammunition, stores and general utility purposes are provided. Six of these have electro-hydraulic Hele-Shaw type motors, and four have direct electric drive. The boat hoisting winches have variable-speed gear electro-hydraulic motors.

A balanced rudder is fitted, operated by a steering gear consisting of four hydraulic cylinders, with single-acting rams, arranged in opposed pairs at each end of the rudder crosshead. The rams are operated by oil under pressure supplied from three variable-speed gear pumps operated by electric motors. Each pump with its motor is contained in a separate watertight compartment, and normally two pumps are in use at one time, the third being held as a standby pump, which can be brought quickly into operation in the event of failure of either or both of the other two, thus ensuring continuity of power supply to the steering gear. In case of complete failure of the

supply of electric power, a variable-speed gear pump, driven by a steam engine, is fitted in the after engine-room. The connecting oil-pressure pipes are arranged for quick change-over in the event of a breakdown of the whole of the main steering power units. With two of the main pumps in use, the gear is capable of moving the rudder from hard-over port to hard-over starboard, and *vice versa*, in 30 seconds, when the vessel is steaming at full speed ahead. This was the first steering gear of its kind to be designed for a British warship, and has proved highly satisfactory in service. About two hundred electric fans of various sizes are used for the general ventilation of the ship, in addition to special fans with brine coolers for the magazine cooling plant. Very great care was exercised in the arrangement of the ventilation, with satisfactory results. The engine-room ventilation is dealt with by four 30-inch exhaust and four 25-inch supply fans, and involved some difficult problems owing to the position of the engine-rooms below the bridge structure.

Turning now to the pumping and flooding arrangements, these are extensive and are designed to deal rapidly with possible damage, and the correction of heel and trim consequent thereon. For compartments other than those occupied by the main machinery, eleven electrically-driven centrifugal pumps, each with an output of 350 tons per hour, are provided. The pumps are of the submersible type and are self-priming. The main and auxiliary circulating pumps are arranged to be available for pumping out their respective compartments and the wings in the vicinity, and in addition a high-capacity steam-driven turbo-pump is fitted in the after boiler-room. For the salt-water service and for dealing with normal drainage and bilge water, nine 50-ton electrically-driven pumps are fitted in those portions of the ship outside the machinery spaces, whilst the fire and bilge steam pumps carry out similar functions inside these spaces.

An exhaustive series of steam trials was carried out on each ship after completion. Following the usual basin trial of the main engines, and the preliminary run at sea, two complete groups of trials were carried out, one with the vessel at about standard displacement, and the other at her deep draught. For each of the two groups of trials, the engines were worked at various powers and runs on the measured mile, at West Looe, Cornwall, were made at each of these powers. During the trial at standard displacement continuous records were made of the oil fuel and water consumptions. *Nelson's* speed at full power at standard displacement was 23.55 knots for 46,000 S.H.P., with a fuel consumption of 0.789 pound per shaft horsepower-hour, while in the deep condition the speed was 23.05 knots for 45,800 shaft horsepower. *Rodney's* maximum speed under similar conditions was 23.8 knots. In addition, steering and circle trials, and stopping, starting, reversing and astern trials were carried out as usual in Admiralty practice. The diameter of the turning circle at full speed was 670 yards, or rather less than three ship-lengths.—"Engineering," March 22, 1929.

SEA TRIALS OF ITALIAN DESTROYERS.*

By Lieut.-Col. F. Dondona, Naval Constructor, R.I.N.

In recent years there have been built by various Italian shipyards a number of types of destroyers, the characteristics of which do not differ greatly, and which have all reached very high speeds—over 36 knots—and represent outstanding examples of modern naval construction. In accordance with the

* Abstract of paper read at the Spring Meetings of the Seventieth Session of the Institution of Naval Architects, March 20, 1929.

conditions of contract, all the groups of similar units have carried out a considerable number of trials (full-speed trials and consumption trials at various speeds), while one ship in each group has also carried out progressive speed trials on the measured mile, so that a number of results of trials are available, with which it has been possible to draw representative curves of the behavior of the ships at different speeds and of the variations of the characteristic factors of propulsion. For two of the types of ships considered it has been possible for the first time in the Italian Royal Navy to carry out the systematic observation of the thrust generated by the propeller at different speeds, by reading the thrust-meters applied to the thrust blocks, so that it has been possible also to draw the curve of thrust powers in terms of speeds; and it has been possible accurately to examine the relation between the H.P. necessary for towing which has been determined in the tank experiments for a certain hull (E.H.P.), and the effective H.P. of the thrust delivered to the ship (T.H.P.), or between the same towing power and the power supplied to the propeller shafts (S.H.P.). If the latter ratio gives us the total propulsive coefficient, which takes account amongst other things of the efficiency of the propeller, it is known that no consideration of the behavior of the propeller has anything to do with the ratio $\frac{\text{E.H.P.}}{\text{T.H.P.}}$—that is to say, with the ratio between the power necessary for towing and the effective thrust. Since the towing power determined in the tank, which has been taken into consideration in the various cases, is referred to the bare hull, *i.e.*, without appendages, whenever there are factors which may similarly be deduced from analogous tank trials, concerning the increase of resistance caused by these appendages, we may deduce the value of the other secondary phenomena which occur in investigating propulsion, *i.e.*, the resistance due to the presence of the propeller giving rise to the phenomenon of the drag (otherwise called thrust deduction) and the action of the wake accompanying the movement of the ship. Another examination rendered possible by the measurement of the thrust is that of the variations of the ratio $\frac{\text{T.H.P.}}{\text{S.H.P.}}$ between the effective power of the thrust and the H.P. transmitted to the propeller. This figure in effect represents the efficiency of the propeller, that is, the ability of the propeller to transform into effective thrust on the ship the power which is transmitted to it by the propeller shaft, multiplied by the mechanical efficiency (which may, however, be considered very near unity) of that part of the shafting between torsion-meter and the propeller.

It is to be observed that the values of the E.H.P. are deduced from the tank experiments, and therefore do not take account, owing to the way in which these experiments are conducted, of anything except the resistance encountered by the model being towed. The values of T.H.P. which have been calculated, taking into account the effective thrust applied to the ship and the effective speed of the ship, represent on their part only the resistance encountered by the moving ship, and are therefore quite comparable with the E.H.P.

On the other hand, the S.H.P. also takes effectively into account the conditions under which the propeller works, so that it can be said that the total propulsive coefficient $\frac{\text{E.H.P.}}{\text{S.H.P.}} = \frac{\text{E.H.P.}}{\text{T.H.P.}} \times \frac{\text{T.H.P.}}{\text{S.H.P.}}$ is divided into one coefficient, $\frac{\text{E.H.P.}}{\text{T.H.P.}}$ which may be called the coefficient of towing, and represents the

greater or lesser capacity of the ship for realizing the conditions of resistance to towing in which the tank trials on the bare hull were carried out, and another coefficient, $\frac{\text{T.H.P.}}{\text{S.H.P.}}$, or propulsive coefficient proper, which represents the greater or less suitability of the propeller for transforming into propulsive energy the energy of rotation which is transmitted to it by the shaft in the particular circumstances presented by the speed of effective advance in the surrounding water and by the effective density of the water in which it works.

The figures of the thrust in the trials of the ships which are being considered, and the ratios $\frac{\text{E.H.P.}}{\text{T.H.P.}}$, $\frac{\text{T.H.P.}}{\text{S.H.P.}}$ and $\frac{\text{E.H.P.}}{\text{S.H.P.}}$ deduced from them which have been plotted in the curves (Fig. 2) permit of following the direction of the various coefficients previously examined, and then of deducing from them conclusions which, supported by other experience of the same kind, enable us to establish a law of a general character or to analyze more clearly a phenomenon which up to now has not been completely elucidated.

The characteristics of the three types of destroyers to which we have referred at the commencement are given in Table I. The ships of the first

TABLE I.

	First Type.		Second Type.		Third Type.	
	Sella.	*Crispi.*	*Sauro.*	*Manin.*	*Turbine Espero.*	*Borea.*
Length at flotation, metres	84·54		90·15		93·20	
Breadth outside of plating, metres	8·63		9·20		9·21	
Mean draught under keel, metres	2·90		3·24		3·27	
Displacement, metric tons	1,150		1,300		1,360	
Wetted surface of hull, sq. metres	802·15		887·5		923·0	
Area of immersed midship section, sq. metres	20·91		23·22		23·46	
Ratio: length to breadth	9·80		9·80		10·12	
Ratio: draught to breadth	0·336		0·353		0·355	
Ratio: length to draught	29·12		27·79		28·52	
Block coefficient	0·532		0·471		0·473	
Prismatic coefficient	0·634		0·605		0·606	
Waterplane area coefficient	0·730		0·735		0·731	
Midship section coefficient	0·837		0·780		0·779	
Pitch of the screw, metres	3·400	3·600	3·300		3·600	3·300
Diameter of the screw, metres	2·900	3·000	3·000		3·000	3·000
Ratio: pitch to diameter	1·172	1·200	1·100		1·200	1·100

type have exactly similar hulls, the only difference existing between the first three (*Sella, Nicotera, Ricasoli*) and the fourth (*Crispi*) being in the propelling machiney. The turbines of the first three are of Parsons geared type, whilst those of the fourth are of Belluzzo type, and for the latter the constructor has considered it advisable to fit different propellers, the particulars of which are given in the table. The four ships of the second type, *Sauro, Battiste, Manin, Nullo,* are all similar, while of the eight ships of the third type, *Turbine* and *Aquilone, Nembo* and *Euro, Borea, Espero, Zeffiro* and *Ostro,* are sisters, except that the *Borea* differs from the others because it has carried out its trials with propellers differing from those of the other ships.

Each of the sixteen ships has carried out a trial of four hours at maximum speed over the measured distance (on a length of about 31 miles for the two *Sauro* and the eight *Turbine,* 24 miles for the two *Manin,* 20 miles for the three *Sella* class, and 8 to 12 miles for the *Crispi*), twice in one direction and twice in the opposite direction, and a consumption trial of six hours at a speed of about 15 knots for the first two types, and of about 20 knots for the third. Moreover, for each group consisting of ships built by separate firms, one ship has carried out trials at progressive speeds, *i.e.,* determining the factors of speed and power for six or seven spots lying between a speed of about 13 knots and the maximum; these trials were carried out over a course of about 3.3 miles (1.3 for the four *Sella* and 2.4 for the two *Manin* class), traversing the course once in each of the two directions for the lower speeds and twice in each direction for the maximum speed. Moreover, trials were carried out at progressive speeds with the *Crispi* at the contract displacement, and with an increased displacement very near that which the ship will reach on service, and progressive speed trials with the *Borea,* which has propellers differing from those of the other units of the group. Consideration has been given to the results of all the trials, *i.e.,* 14 for the three *Sella,* 11 for the *Crispi* at contract displacement, 16 for the two *Sauro,* 14 for the two *Manin,* 10 for the two *Turbine,* 11 for the two *Nembo,* 15 for the three *Espero,* and 8 for the *Borea*—99 in all.

The first attempts to plot as curves the results from all the similar units of each type, even when built by different firms, has shown the impossibility of arriving at positive results. It seemed, therefore, to be more satisfactory to collect into separate curves the results of trials of similar ships built by any one firm. In this way it was possible to have more homogeneous data available, and to plot sufficiently accurate curves representing a fair mean of the various factors from the trials. Curves have been plotted representing as a function of the speed of the ship V in knots:—

(*a*) The number of revolutions per minute, or n;

(*b*) the percentage slip, $\frac{60\,n\,p - 1{,}852\,V}{60\,n\,p}$;

(*c*) the number of revolutions per minute per knot, $\frac{n}{V}$;

(*d*) the power read from the torsion-meter in shaft horsepower, S.H.P.;

(*e*) the power deduced from the readings of the thrust at the thrust-meter in horsepower, T.H.P.;

(*f*) the towing power deduced from the Tank trial curve for the bare hull (*i.e.,* without appendages) in horsepower, E.H.P.;

(*g*) the coefficient of towing, $\frac{\text{E.H.P.}}{\text{T.H.P.}}$;

(*h*) the coefficient of propulsion, $\frac{\text{T.H.P.}}{\text{S.H.P.}}$;

(*i*) the total propulsive coefficient, $\frac{\text{E.H.P.}}{\text{S.H.P.}}$;

(*l*) the value of the Admiralty constant, $C = \frac{D^{\frac{2}{3}} V^3}{\text{S.H.P.}}$

For an easier and more comprehensive reading it was considered convenient to take part of these curves in another form as diagrams to be read as a function of the number of revolutions per minute. In these diagrams have also been added the curves representing:—

(*m*) the value of the Admiralty constant referred to the power for towing in the Tank;

$$C_e = \frac{D^{\frac{2}{3}} V^3}{\text{E.H.P.}}.$$

(*n*) the value of the Admiralty constant referred to the power deduced from the reading of the thrust at the thrust-meter:—

$$C_T = \frac{D^{\frac{2}{3}} V^3}{\text{T.H.P.}}.$$

(Of the 14 diagrams in the paper, those for the *Borea* are reproduced in Figs. 1 and 2.)

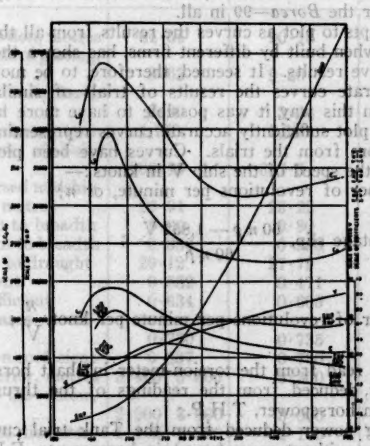


Fig. 1.—Typical Admiralty Constant Curves for Italian Destroyers.

TABLE III.

	First Type.		Second Type.		Third Type.		
	Sella.	*Crispi.*	*Sauro.*	*Manin.*	*Turbine.*	*Espero.*	*Borea.*
E.H.P. / S.H.P. maximum	0·528	0·531	0·570	0·553	0·496	(0·637)?	0·410
Corresponding speed, knots	29	21·2—33·5	23·9	21	20·8	(20·6)?	26·2
Corresponding revolutions	312	210—350	235	216	270	(193)?	290
E.H.P. / T.H.P. maximum	—	—	0·670	0·615	0·600	—	0·505
Corresponding speed, knots	—	—	21·7	22·8	20·0	—	18·3
Corresponding revolutions	—	—	221	240	185	—	184
T.H.P. / S.H.P. maximum	—	—	0·660	0·710	0·623	—	0·590
Corresponding speed, knots	—	—	23·6	18	20—32	—	23·5
Corresponding revolutions	—	—	260	180	287—345	—	250
C_s minimum	281·5	281·5	295	295	312	312	312
Critical speed for voyage, knots	29·2	29·2	30·4	30·4	30	30	30
C_t minimum	—	—	225	231	231	—	225
Critical speed for thrust, knots	—	—	33·4	32·8	23·3	—	32·3
Corresponding revolutions	—	—	300	330	365	—	400
C minimum	148	150	139	<122	143	124	104
Critical speed for propulsion, knots	30·6	29·2	35·8	>38	34·3	36	38
Corresponding revolutions	350	300	455	>500	382	408	525
Data referring to maximum speed of group :—							
Maximum speed, knots	38·06	38·6	37	37·4	39·5	38·4	36·55
Corresponding revolutions	463	415	490	488	496	448	491
C_s	—	—	258	258	278·5	—	239
C	152	171·5	139·2	122·6	148·2	125·5	104·6
E.H.P. / S.H.P.	0·470	0·383	0·449	0·390	0·420	0·300	0·312
E.H.P. / T.H.P.	—	—	0·795	0·697	0·722	—	0·710
T.H.P. / S.H.P.	—	—	0·560	0·471	0·532	—	0·436
V / E.H.P.							

S.H.P.													
†I.H.P.	—	—	—	—	—	—	—	—	0·785	0·807	0·722	—	0·710
‡I.H.P.													
§I.H.P.	—	—	—	—	—	—	—	—	0·560	0·471	0·522	—	0·426
$c = \frac{V}{\sqrt{L}}$ for $\frac{\text{S.H.P.}}{\text{S.H.P.}}$ maximum								3·16	2·21—3·06	2·615	2·21	2·765	(2·25)?	2·715
$c = \frac{V}{\sqrt{L}}$ for V maximum								4·15	4·21	3·9	3·94	4·03	3·88	3·79

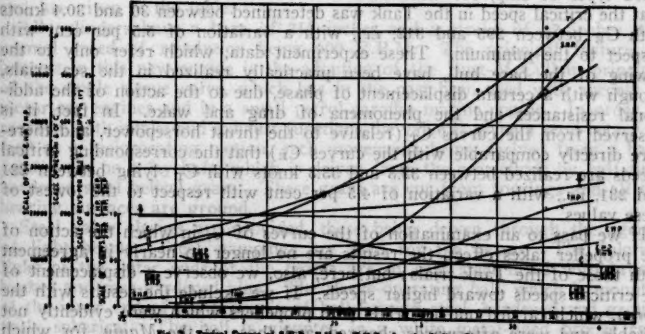


Fig. 2.—Typical Curves of S.H.P., E.H.P., T.S.P., &c., for Italian Destroyers.

Proceeding now to analyze the results with the aid of the diagrams, some observations can be made which, however, should be preceded by an examination of the characteristic elements of the different hulls at the constant displacement of 1200 tons considered as the point of reference of all the powers examined (Table II).

TABLE II.

	First Type.	Second Type.	Third Type.
Length at flotation, m.	84·67	90·16	93·20
Breadth outside of plating, m.	8·63	9·20	9·20
Mean draught under keel, m.	2·99	3·08	3·02
Displacement, metric tons	1,200	1,200	1,200
Wetted surface of the hull, sq. m.	817·6	863·5	873·85
Area of immersed midship section, sq. m.	21·69	21·70	21·16
Ratio : length to breadth	9·81	9·80	10·13
Ratio : draught to breadth	0·360	0·335	0·327
Ratio : length to draught	28·36	29·28	30·89
Block coefficient	0·537	0·488	0·452
Prismatic coefficient	0·637	0·598	0·593
Waterplane area coefficient	0·733	0·728	0·723
Midship section coefficient	0·842	0·766	0·762

In Table III, in order to give a clearer view, the more salient factors deduced from the curves have been collected together.

Some observations may be suggested by an examination of the direction of the curves C, C_T C_E, which show the way in which with the different hulls it was best to utilize the power of the propelling machinery at the different speeds. The curve C_E shows that the hulls, as designed, have all reached a minimum C_E, and therefore a maximum relative power at fairly uniform speeds lying between 29.2 and 30.4 knots. Considering only the second and

third types of ship, for which we have more complete trial data, we see that the critical speed in the Tank was determined between 30 and 30.4 knots with C_R between 295 and 312, *i.e.*, with a variation of 5.5 per cent with respect to the minimum. These experiment data, which refer only to the towing of the bare hull, have been practically realized in the sea trials, though with a certain displacement of phase, due to the action of the additional resistances and the phenomena of drag and wake. In fact, it is observed from the curves C_T (relative to the thrust horsepower, and therefore directly comparable with the curves C_R) that the corresponding critical speeds are realized between 32.3 and 33.3 knots with C_T lying between 221 and 231, *i.e.*, with a variation of 4.5 per cent with respect to the lowest of these values.

If we pass to an examination of the curves of C, in which the action of the propeller takes effect, the results are no longer so nearly in agreement with those of the Tank trials; but here, also, we observe a displacement of the critical speeds toward higher speeds. If we exclude the results with the *Borea*, which carried out her trials with propellers which were evidently not suitable, and were afterwards changed, and those of the *Manin*, for which the critical speed appears abnormal because it passes out of the field of the experiments, it is seen that for the other units the critical speed is found between 34.3 and 36 knots with C lying between 124 and 143, or with a variation of 15 per cent, with respect to the lowest of these values.

Finally, it is not without interest to remark the figure at which, for the ships considered, the value of a in the formula $V = a\sqrt{L}$ in Table III is maintained. We observe that a varies from a minimum of 3.79 to a maximum of 4.21, with a mean of 4, which represents a good mean value for modern ships of this type. A characteristic phenomenon, to which the attention of students of naval architecture should be called, is given by the direction of the curves representing the T.H.P., which have been plotted from the formula $\text{T.H.P.} = \frac{\text{Thrust in kg.} \times 1{,}852\,V}{3{,}600 \times 75}$

The last part of these curves, in the neighborhood of the maximum speed reached by the ship, has an evident tendency to droop and maintain itself parallel to the axis of the abscissæ. If the curve were to assume a decidedly horizontal direction, it would signify that there was, in effect, a diminution of the thrust, which would tend to compensate for the increase of speed. But this would be an absurdity, and, in fact, it does not actually happen; it may, however, be said that the thrust has a tendency to remain constant so that the T.H.P. values increase only because of the increase of speed. It is extremely interesting that this phenomenon occurs with all the ships on which it was possible to make exact measurements of the thrust, because it may lead to some unforeseen explanation of the phenomenon of propulsion at high speeds.—"Shipbuilding and Shipping Record," March 28, 1929.

RICARDO-BROTHERHOOD HIGH-SPEED DIESEL ENGINE.

Brief reference was made in our last issue to the new single-sleeve valve four-stroke cycle high-speed airless-injection Diesel engine which Peter Brotherhood Limited, Peterborough, have produced in collaboration with H. R. Ricardo & Co. The engine is built to the designs of Mr. Ricardo's firm, the actual manufacture and marketing of the unit being left in the hands of Peter Brotherhood Limited. At the present stage of development it is proposed to standardize one size of cylinder, giving 50 B.H.P. per line.

The four sizes of engine, together with their principal dimensions, weights, etc., are given in the table.

All cylinders have a bore of 7½ inches and a stroke of 12 inches, the normal speed of revolution being 900 R.P.M. The principal feature of interest in this new engine is the adoption of the Burt-McCullam principle of the single sleeve valve, which has in the past been used with success in motor-car engines and in a small petrol-paraffin marine engine. In order to perform both intake and exhaust functions the valve is given a semi-rotary as well as a reciprocating motion, such movements being derived from a ball-jointed pin on a layshaft, which is driven from the crankshaft by half-time hardened steel helical gears. These gears are cut from forgings and all the bearing surfaces are ground.

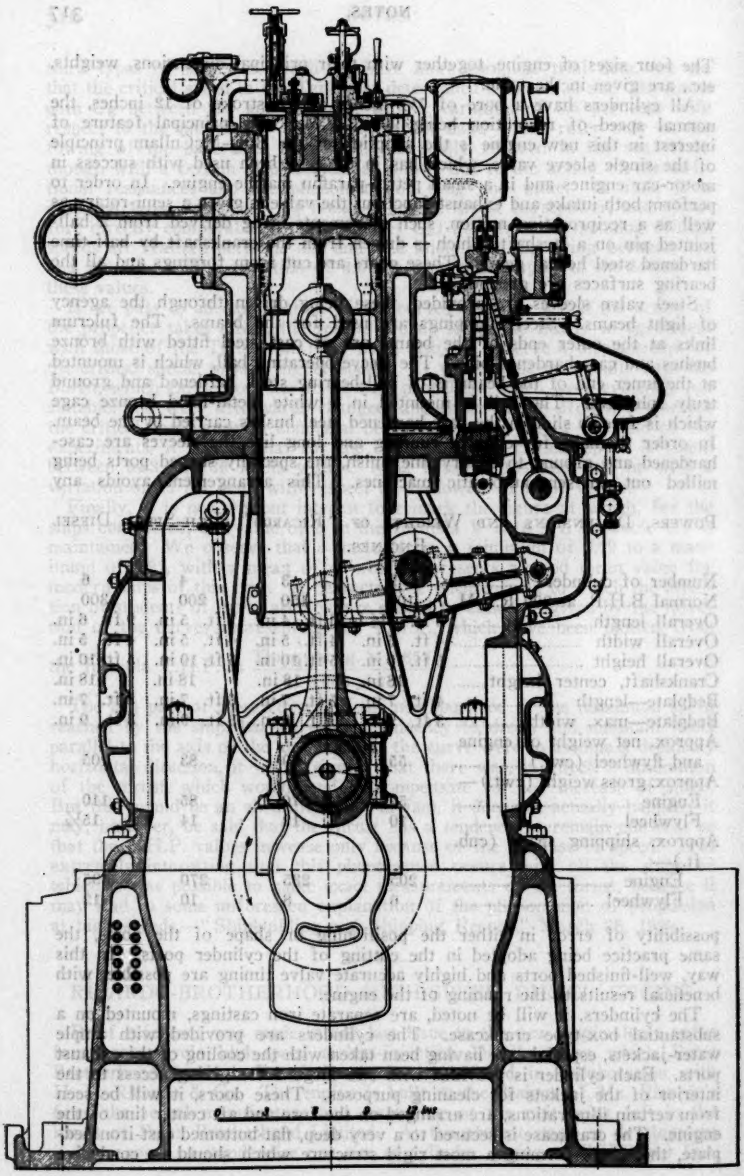
Steel valve sleeves are provided, these being driven through the agency of light beams. Steel stampings are used for the beams. The fulcrum links at the outer ends of the beams are of cast steel fitted with bronze bushes and case-hardened pins. The sleeve-operating ball, which is mounted at the inner end of the beam, is of ball-bearing steel, hardened and ground truly spherical. The ball is mounted in a white metal-lined bronze cage which is free to slide in the case-hardened steel bushes carried by the beam. In order to make for efficient running and long life the sleeves are case-hardened and ground to a very fine finish, the specially shaped ports being milled out on semi-automatic machines. This arrangement avoids any

Powers, Dimensions and Weights of "Ricardo" High-Speed Diesel Engines.

Number of cylinders	2	3	4	6
Normal B.H.P. at 900 R.P.M.	100	150	200	300
Overall length	5 ft. 3½ in.	6 ft. 4 in.	7 ft. 5 in.	9 ft. 6 in.
Overall width	4 ft. 5 in.	4 ft. 5 in.	4 ft. 5 in.	4 ft. 5 in.
Overall height	5 ft. 10 in.	5 ft. 10 in.	5 ft. 10 in.	5 ft. 10 in.
Crankshaft, center height	18 in.	18 in.	18 in.	18 in.
Bedplate—length	4 ft. 7 in.	5 ft. 7 in.	6 ft. 7 in.	8 ft. 7 in.
Bedplate—max. width	3 ft. 9 in.	3 ft. 9 in.	3 ft. 9 in.	3 ft. 9 in.
Approx. net weight of engine and flywheel (cwt.)	55	69	83	105
Approx. gross weight (cwt.)—				
Engine	57	70	85	110
Flywheel	10	12	14	15½
Approx. shipping space (cub. ft.)—				
Engine	205	235	270	335
Flywheel	6	8	10	12

possibility of error in either the positioning or shape of the ports, the same practice being adopted in the cutting of the cylinder ports. In this way, well-finished ports and highly accurate valve timing are possible, with beneficial results to the running of the engine.

The cylinders, it will be noted, are separate iron castings, mounted on a substantial box-type crankcase. The cylinders are provided with ample water-jackets, especial care having been taken with the cooling of the exhaust ports. Each cylinder is provided with two large doors giving access to the interior of the jackets for cleaning purposes. These doors, it will be seen from certain illustrations, are arranged on the fore and aft center line of the engine. The crankcase is secured to a very deep, flat-bottomed cast-iron bed-plate, the whole forming a most rigid structure which should be conducive



Section through Cylinder of Ricardo-Brotherhood Single-Sleeve Valve High-Speed Diesel Engine.

to steady, vibrationless running. The bedplate is water-cooled by jackets running along each side of the crank pits, while the main bearings of the crankshaft are carried by double webs which form stiff box section girders across the bedplate in way of each journal. The bearing caps are steel stampings, each secured by four collar studs.

The pistons are of aluminum alloy and are provided with five pressure rings and one scraper ring. The use of uncooled aluminum alloy pistons makes for a reduction of reciprocating weight and so of inertia forces, while it is claimed to lead to material economy in the consumption of lubricating oil, as well as reducing the temperature of the piston crowns. The connecting rods are hollow alloy steel stampings, and are machined all over. The small end of the connecting rod is provided with a bronze bush for the large hollow gudgeon pin, which is case-hardened and ground and is free to float in the rod end and in the piston.

It is worthy of note that the crankshaft is of the built-up type, an unusual feature for a small high-speed oil engine. The journals are shrunk into the crank webs, while the crank pins are held in place by means of substantial pinch bolts. This construction has the advantage that the pins and journals may be case-hardened, while it also allows of their easy and economical replacement. The big-end construction is rather interesting inasmuch as the loose bronze big-end bush is free to rotate on the crank pin. This bearing is contained in a case-hardened steel bush carried in the eye of the connecting rod. Relative motion between the bronze bearing and the bush and the bronze bearing and the crank pin enables the rubbing velocity at the pin to be reasonably low, an important factor in high-speed engine design. The main crankshaft bearings, which are of generous area, consist of white metal-lined bronze shells.

The forced lubrication system is a very thorough one. From an oil supply reservoir cast in the bedplate, lubricating oil is drawn by a pump and supplied under pressure to the journals, big end bearings, layshaft and sleeve-operating gear. The pump suction is surrounded by a gauze filter of ample size, while before the oil is passed into the supply pipes it is circulated through an Auto-Klean strainer. After circulation around the engine the oil falls to the sump, where it is picked up by a second pump, passed through cooling pipes in the bedplate water jacket, and returned to the supply reservoir. The lubrication system, it will thus be seen, is entirely self-contained, no extraneous pumps, filters or coolers being required. A hand pump is provided for priming the lubrication system before starting up the engine.

Fuel injection is on the spill valve principle with automatic fuel valves, a separate fuel pump being provided for each cylinder. The pump plungers are of hardened steel, while renewable hard cast-iron pump liners are provided in the pump block. The fuel pumps are driven from a connecting link and crosshead through the fulcrum lever at the outer end of the sleeve-valve operating beam. Mechanically-operated suction and spill valves are provided, governing of the engine being effected by varying the point of opening of the spill valve, thus leaving the fuel injection timing unaltered at all loads. A centrifugal governor is provided.

The automatic fuel valves are of interest in that the usual needle is replaced by a spring-loaded ball valve. The sprayer nozzle cap is of hardened steel with a single large-diameter hole in its center. The shape of the combustion chamber and the arrangement of the ports is such that a high degree of turbulence is given to the air charge, this being relied upon, in association with rapid closing characteristics of the spill valve system, to effect efficient combustion. The simple, single-hole sprayer used has only been adopted after exhaustive experiments, it having been found to admirably suit

the cylinder conditions and combustion chamber shape which obtain in this engine. The cylinder heads, in passing, are thin steel forgings, machined all over, the combustion chamber being completely surrounded by water.

The method of starting the engine is interesting. Instead of the usual cylinder starting-air valve a small four-cylinder, single-acting, poppet valve, water-cooled compressed-air motor is placed alongside the flywheel of the six-cylinder engine and generator. The pinion on the end of the crankshaft to this engine is of the well-known Bendix self-releasing type, such as is used in motor-car starter gears. Starting air at a pressure of 250 pounds per square inch is used, a water-cooled air compressor which is built into the engine and driven at reduced speed by the sleeve-operating helical gears, being used to provide starting air. This compressor can be quickly put into action when required by means of a friction clutch operated by a conveniently placed lever.—"The Marine Engineer and Motorship Builder," March, 1929.

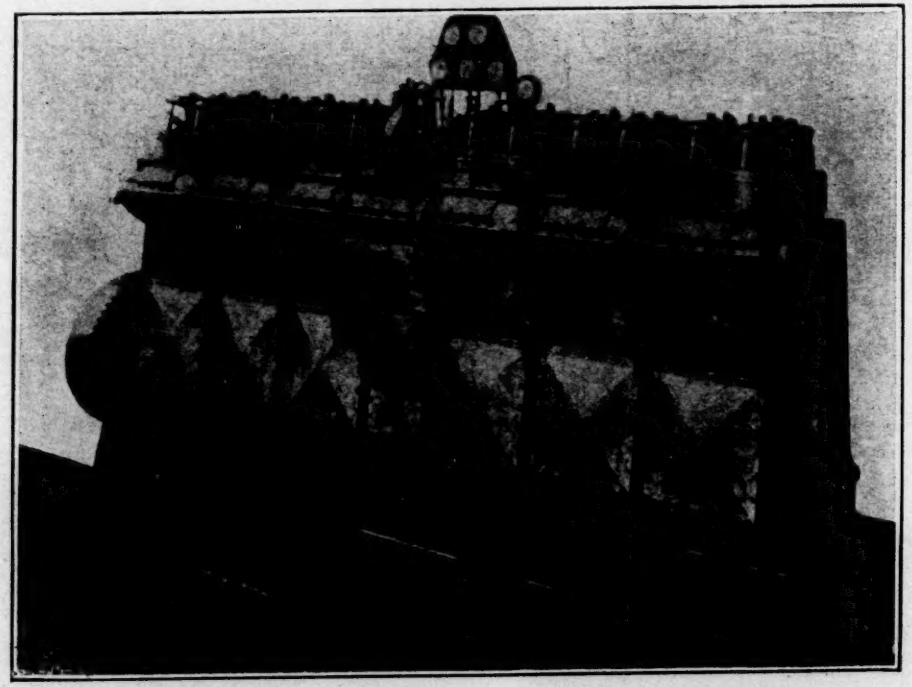
A SIX-CYLINDER 750 B.H.P. LIGHT WEIGHT OIL ENGINE.

In the course of a visit paid to the Naval Construction Works of Vickers-Armstrongs, Ltd., at Barrow-in-Furness, we recently saw in course of construction and on the test bed, examples of a new type of high-speed light-weight oil engine, which we show in the accompanying engravings. The design is noteworthy for the care which has been taken in reducing the weight of parts without sacrificing essential strength. The new engine is one which the makers state is suitable both for the smaller class of submarines and for auxiliary lighting sets, while it can possibly be employed for oil engine drives on locomotives of the oil or oil-electric type. The engine is non-reversing and the absence of reversing equipment introduces additional simplicity, particularly with regard to the cam shaft. It is a six-cylinder four-stroke unit, with the fuel pump and the controls placed at the center of the engine.

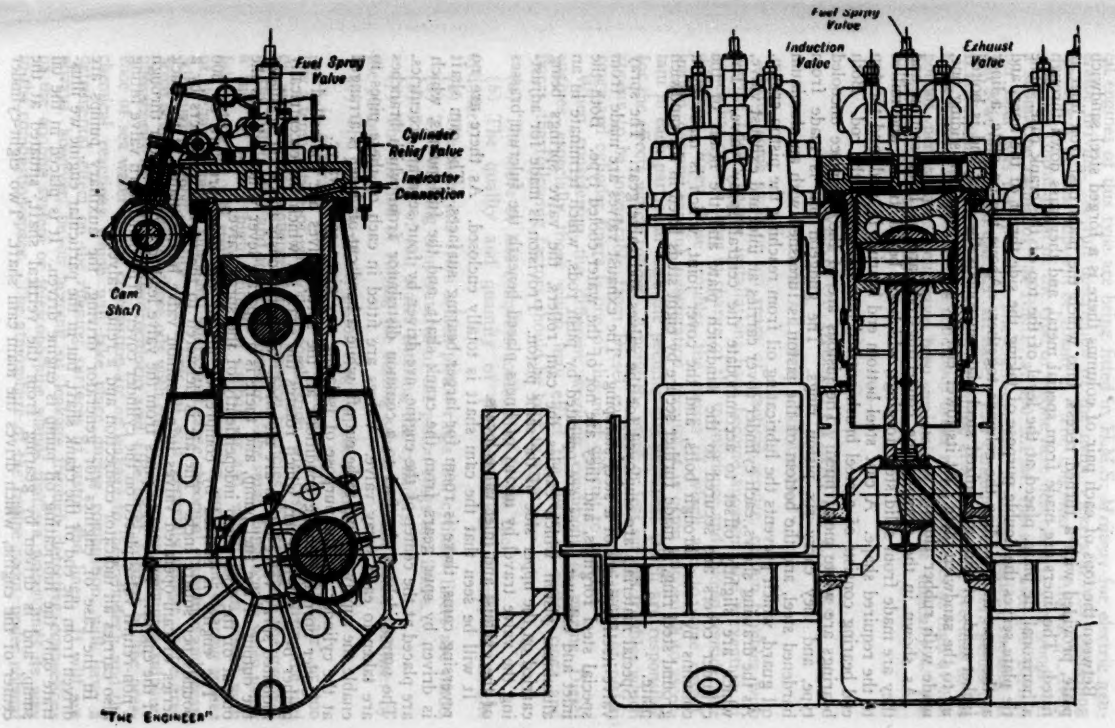
The cylinders have a bore of 14 inches with a stroke of 14.8 inches, and at a full power submarine rating the output is 750 B.H.P. at 450 R.P.M. When employed as a continuously running engine for electric power, 300 Kw. may be developed at a running speed of 350 R.P.M., that rating being such that a 25 per cent overload is then available over long periods.

The view reproduced shows the external appearance of the engine, which is clean and neat, while two typical sections through the cylinder and crank shaft are shown in the accompanying drawing. The type of bedplate may be modified to suit the conditions of service, and we show a bedplate of the submarine type, which is made in cast steel. It is light in section and is strengthened by well-placed ribs. For dynamo driving, a deeper form of bedplate, made either in cast iron or cast steel, and furnished with an oil sump, is employed. The engine weight will vary, of course, slightly with the type of bedplate adopted, but the submarine type of engine illustrated works out, we are informed, to about 13.7 tons, or at the above-mentioned rating about 40 pounds per B.H.P.

In order to obtain such a saving in weight, considerable care has been exercised in the design of detail parts. We have already referred to the light ribbed steel bedplate. The crank shaft is made in two halves, and is forged solid in oil-toughened steel. All the main bearings consist of steel shells fitted with white metal linings. As will be observed, the cast steel columns, which are of light ribbed section, straddle the main bearings.



750 B.H.P. Four-Stroke High-Speed Oil Engine.
Vickers-Armstrongs, Ltd., Barrow-in-Furness, Engineers.



Sections of Engine.

Between the tops of each pair of columns there is a forged steel sandwich plate, provided with a turned recess, into which the plain cast iron liner is fitted. The liners are made from special metal, and they pass down through a horizonal tie plate, placed at the level of the top of the crank case. The tie plate serves the double purpose of taking the side thrust of the liner, and at the same time binding the columns together. Above the tie plate, a liner jacket made of thin galvanized steel plate surrounds the liner and is bolted up to the sandwich plate. At its lower end the jacket joint is a sliding one, made with rubber rings.

As shown in the drawing the connecting-rods are of light section and they are made from solid steel forgings, which are bored, turned and milled to the required shape. A cast steel bottom end bearing is used, and the top end bearing consists of a steel brush pressed into the eye of the rod. Both bearings are white metal lined. The pistons are of the one-piece uncooled type, and they carry six piston rings. The gudgeon pin is made from hardened steel, and the bottom of the piston is furnished with an aluminum oil guard, which prevents the lubricating oil from reaching the piston crown. As the drawing shows, each cylinder cover carries an inlet and exhaust valve, which are slightly offset to accommodate the central spray valve. The cylinder covers are secured to the sandwich plate and the heads of the columns by four through bolts, and the cover joint, which is made by a ground steel ring, is made further secure by eight studs fixed in the sandwich plate.

Special materials are employed for the valves and valve gear. The spray valve itself consists of a steel stamping. The exhaust valves are made from special steel forgings, and they are not of the water-cooled type. Both the inlet and exhaust valves are operated by push rods, which terminate in an aluminum piston which carries the cam rollers, the valve springs being carried on the upper side of the same piston. Provision is made for adjusting the valve travel by means of liners placed beneath the fulcrum brasses of the exhaust and inlet valve levers.

It will be seen that the cam shaft is totally enclosed. As there are no reversing cams, there is room for larger bearing surfaces. The cam shaft is driven by spur gears from the crank shaft, and the fuel pumps, which are placed at the center of the engine, are driven by four separate eccentrics. The pumps deliver the fuel to a common distributor from which branches are taken to each spray valve. Valves are fitted in each of these pipes to enable the supply of fuel to be cut off, while a hand cut-off lever is arranged at the cylinder for the purpose of isolating any given spray valve.

Under normal running conditions all the spray valves are controlled together by a single lever, which rotates the shaft on which the eccentrically mounted spray valve levers are mounted. Simultaneously this lever regulates the output of the fuel pump, and there is a separate lever which adjusts the output of the fuel pump independently of the spray valve.

The engine is started by compressed air at the usual pressure of 600 pounds per square inch. The air is delivered to the various cylinders by a series of cam-operated valves in a common valve box, situated at the back of the engine. The supply pipes from the valve lead to each cylinder through a non-return valve placed on the cylinder cover. The non-return valve fitting also carries an indicator connection and a relief valve.

In the case of engines for generator driving, the auxiliary pumps are driven from the end of the crank shaft, but in the particular engine we illustrate only the lubricating oil pump is engine driven. It is placed in the oil sump, and is driven by gearing from the vertical shaft, situated at the center of the engine, which drives the main cam shaft. Two eight-cylinder

oil engines of this type are being fitted in H.M.S. *Medway* for electric generator driving. A full description of the main Vickers-M.A.N. propelling machinery of this ship will be found in "The Engineer" of August 31, 1928. The fuel consumption of the type of engine here described is very good. At a submarine rating of 750 B.H.P. at 450 R.P.M. the fuel used per B.H.P. per hour was 0.42 pound. The eight-cylinder *Medway* engines running at 350 Kw. output and 350 R.P.M. have a fuel consumption of 0.375 pound per B.H.P. hour, and 0.397 pound when running at a 50 per cent overload at the same speed. Under overload conditions brake mean pressures as high as 95.5 pounds per square inch have been maintained without difficulty.—"The Engineer," Feb. 15, 1929.

THE WEAR OF DIESEL CYLINDER LINERS.

(From a Dutch Correspondent.)

After some eight or ten years of practical service, certain troubles with marine Diesel engine installations have become apparent, the prevention of which has caused those in charge of the maintenance and upkeep considerable difficulty. Even those troubles, the causes of which are, to some extent, known, demand careful investigation, because it is often impossible to ascertain in each particular case the peculiar circumstances which are responsible for the effects which follow. Among the more or less serious troubles which appear to be inseparable from Diesel engine operation, the wear of piston rings and cylinder liners is one of the most difficult to prevent.

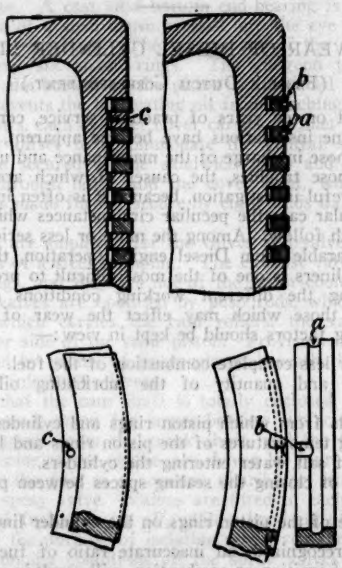
When considering the different working conditions of marine Diesel engines, especially those which may effect the wear of piston rings and liners, the following factors should be kept in view:—

(*a*) The more or less complete combustion of the fuel.
(*b*) The quality and quantity of the lubricating oil supplied to the cylinders.
(*c*) The materials from which piston rings and cylinder liners are made.
(*d*) The working temperatures of the piston rings and liners.
(*e*) Possibility of salt water entering the cylinders.
(*f*) The method of closing the sealing spaces between piston and liner by the piston rings.
(*g*) The pressure of the piston rings on the cylinder liner.

As is generally recognized, an inaccurate ratio of fuel and air, or too low a temperature of ignition or combustion, will result in carbonizing the fuel oil, with the consequence that the carbon particles will penetrate the grooves of the piston rings and act as an abrasive. The same thing occurs when using a fuel oil with a high content of asphalt or ash, which, when mixed with lubricating oil, will stick to or fill up the working surface and clearing spaces of the piston, piston rings and liners, and cause abnormal resistance and wear. Although a good quality lubricating oil will generally appear to be free from acid, the contact of these oils with combustion gases containing sulphur, phosphorus or eventually hydrochloric acid (the latter formed by the presence of sea water in the combustion-air or leakage to the cylinder), and the chemical reactions which will take place during combustion, will have a detrimental effect on the running surfaces of piston and liner. It will not always, however, be on account of leakage through defective connections of the cooling system or telescope pipe glands that sea water finds its way to the combustion space. The atmosphere of the engine room, under normal

conditions at sea, and particluarly with stormy weather, is more or less mixed up with very fine sea water particles, which find their way to the air intake pipes of the engines. It is, of course, impossible wholly to prevent cylinder liner wear due to this cause, but without doubt it could be reduced by furnishing the air-suction pipes, when leading to the open deck, with a moisture-separating device.

The difference in the phenomenon of cylinder wear in the case of four-stroke and two-stroke cycle cylinder liners is interesting. Wear at the top end of the four-stroke liner decreases very rapidly during the first part of the down stroke and diminishes in a slight proportional direction to the bottom, whereas, in two-stroke cycle engines, the maximum wear extends



METHODS OF REDUCING PRESSURE BEHIND THE RINGS IN FOUR-STROKE AND TWO-STROKE CYCLE ENGINES.

practically over the whole length of the liner above the exhaust ports, and is in accordance with the last-named supposition. The intake-air of two-stroke cycle engines, containing salt-laden moisture, sweeps the liner from the scavenge ports up to the cylinder cover and down to the exhaust ports during the scavenging period, in which period the salt ingredients come into more intimate contact with the cylinder liner than when entering the cylinder by the suction air valves in one direction in the case of a four-stroke cycle engine.

The rate of wear, either by mechanical or by chemical causes, will increase with the temperature of the bearing surfaces and the normal pressures which they are called upon to withstand. In this connection the effective

cooling of piston and liner will reduce the wear, whereas a good fit of the piston rings in their grooves and a normal clearance space between piston and liner will favor the heat-flow from the piston, and through the liner to the cooling water. As the main cause of cylinder wear, in the writer's opinion, lies in the pressure of gases behind the piston rings, it is obvious that the best remedy against this evil will be to reduce the pressure behind the rings as much as possible.

Generally, six to eight piston rings, with one scraper-ring, are fitted to a Diesel engine piston, the two or three top ones being exposed to the maximum combustion temperatures and pressures, and consequently these top rings will cause and have the lion's share of the overall wear. Not only will the top and bottom surfaces of rings and grooves suffer from the pressures in an axial direction, assisted by the radial movement of the rings in their grooves by a less accurate alignment of the piston in the cylinder. The clearance of the rings in the grooves in an axial direction, and the opening at their joints, will facilitate the entrance of high-pressure gases to the back side of the rings. To reduce this gas pressure behind the top-piston rings in a four-stroke cycle Diesel engine, the bottom side of the grooves were brought into communication with each other, and with the atmosphere, by boring three or four 3/16-inch through holes *c*, as shown in the left-hand sketch, in the circumference of the piston body, by which measure the wear of the rings and liner diminished considerably. In the case of a two-stroke cycle engine the same purpose was served by providing the first and second rings with a circumferential groove *a* (right-hand figure) and bringing the groove in communication with the combustion space by three or four slots *b;* the slots *b* terminated on about 1 inch distance from the opening of the joints of the rings. By this measure the gas pressure behind the rings was counterbalanced partly by the gas pressure in the grooves *a* between the rings and the cylinder liner, with favorable results.—"The Marine Engineer and Motorship Builder," March, 1929.

THE UNPRACTICAL MAN.

At the convivial gatherings, which are interspersed through the long winter among the more serious proceedings of the learned and technical societies favored by the engineering profession, it is the accepted custom that a little harmless adulation should be indulged in. Thus, the civil engineers are told that without them would be impossible either the fulfilment of our needs in peace, or the success of our forces in war. The mechanical men are flattered by the fact that upon their shoulders falls the responsibility of making it feasible for all other branches of the profession to carry out their ideas. The naval architect, without whose assistance life in this island would be possible only for a few of us, is reminded that his calling is at least as old as Noah—perhaps discoveries at Ur will carry him back before long to antideluvian times. Each and all, according to the success which has attended the choice of the principal guest of the evening, go home that night with a mild sense of elation that they undoubtedly are *the* people, and so to bed.

By a strange throw of fate two particular societies were last week to be found in somewhat curious juxtaposition in this way. One of these bodies was the Institution of Mining and Metallurgy, whose always enjoyable annual celebration was held on Wednesday, the 10th inst. The Institution claims that of its members more than 60 per cent are engaged abroad, but the hospitality of the body is such that the proportion of those present at the dinner, attending as guests, must have easily equalled that figure. The

gathering was quite impressive. This, of course, is as it should be, for whatever may be claimed by other institutions, mining precedes all engineering work, and next comes metallurgy—after which we others, civils, mechanicals, etc., put to use what our brethren have thus provided for us. If ever praise were due, therefore, we take it it would be justified on such an evening, and we are glad to say it was not lacking, though very weighty words from Sir Halford Mackinder tended to check undue exhilaration. The future, as Sir Halford sees it, is fraught with dangerous possibilities, from the monopolistic possession by various nations of certain mineral resources, upon the development of which, in one direction or another, will turn the chances of world peace or war.

The second body to which we have referred was the Institution of Engineers-in-Charge, whose unique function was held on Friday last, and whose gathering was also graced by a number of distinguished guests. Again, it was natural that the hosts should be the subject of a little compliment, for without men of their type the wheels of the world would, of course, cease to revolve. The question was asked by Mr. J. Wardlaw Milne: Engineers in charge—of what? To which he said he could only find one reply: of everything. With this we may well agree, for, though we are not too certain of the regulations for admission to the ranks of this particular body, the term is not their particular monopoly. The official responsible for the reconstruction of the Welland Ship Canal, now being described in our columns, is designated "Engineer in Charge," and thus from such great works downwards, this title finds appropriate application. As Sir Brodie Henderson stated on Friday, the term is one designating practical men, or as Dr. Ormandy put it a little later in the evening, the class whose duty it was to make the things contrived by a Sir Brodie Henderson or a Sir Henry Fowler, perform their functions satisfactorily.

Our readers will possibly wonder why it should be considered curious that these functions followed one another closely. The reason is to be found in this: At the second evening, as we have suggested, the general theme was the value to the world of the practical man; while at the earlier function, Professor S. J. Truscott, the president, had laid stress upon the urgent need of superseding the practical man by scientific supervision. The curious coincidence of the exposition of these two opposing points of view must have struck not a few who were privileged to be present at both functions, while it impressed itself upon us the more because Professor Truscott had, not long before, explained his views to us upon this very point. As we follow Professor Truscott, he considers that what he calls the "practical man" has been the bane of mining for many years, and no less the curse of our industries at home. According to our informant, the foreman type is in complete control of our engineering works. When there is the possibility of a job, the information is taken down to a practical man in the shops, who scratches his head a bit, buries a bit of wood in the sand, and then just sees what turns out. Usually, we understand, the part of whatever it may be is vastly too heavy, and Germany gets the order. This is a very different picture from that drawn by Sir Brodie Henderson, who, on Friday last, spoke of the practical man as the one who could so best combine theory and practice as to accomplish some specific purpose with the least waste, and this we think more nearly represents the normal view. In fact, the more one looks into Professor Truscott's contentions, the more one is driven to the conclusion that he is, wittingly or not, misapplying the term. The alternative to the practical man is by no means, as Professor Truscott would seem to argue, the scientific man; it is the *unpractical* man, and from this heaven help us, whether science be thrown in as a make-weight, or not.

Engineers in charge of our industries will possibly find some difficulty in recognizing themselves in Professor Truscott's picture, and equally we believe would the artist fail to find his model if he actually visited works in search of it, for he would be looking for little more than an overgrown type of skilled artisan. No one will deny that such a class at one time existed and indeed did much to place our work in the front rank; all honor to them. In their day education was hard to come by, but it has now for a long while been properly stressed, though the fact remains that the leading institutions still lay great store by practical training, for fear lest the unpractical theorist should gain the ascendancy.

Responding to the toast of "The Mineral and Metal Industries" on the evening on which Professor Truscott occupied the chair, Sir Godfrey Fell quoted *Stalky and Company*. Now, the verses in the opening pages of *Stalky* are pre-eminently concerned with the men who quietly run things, "diligent in all they do, keen in their vocation." It is undoubtedly the praises of the practical man on the job that Kipling sings in these lines, yet, a few moments later the assemblage was being told that the practical man should give place to "scientific supervision." The fact is that, rightly or wrongly, Professor Truscott has acquired a somewhat different view from the rest of us of what the term implies, and, in medical parlance, possibly a little corrective would not be amiss.

Supposing we did train all our young men in the ways of super-scientific supervision, and steeped our draughtsmen in theory, what should we do with them when it came to the practical question of bread and butter? Everybody cannot find occupation in designing, else who would do the making—or, if made, who would do the running, and after running who would do the repair work? Thus, while granting that first-class design is essential, we still hold that it takes all kinds to make a world, and in *our* world the various types of engineer are mostly complementary to each other. Mining and metallurgy would not get far without the rest, and the fact remains that in these branches, too, as in others, the practical man, as we understand him, is not badly represented. Sixty per cent of its membership abroad is not a bad record for the Institution, and we may be sure that in this category are included many an one who is bearing in a very practical manner the heat and burden of the day, facing imperturbably the many difficulties which beset the path of mining enterprise, and coming triumphant out of situations which would simply overwhelm the "scientific supervisor." Though not of the breed who win knighthoods for spectacular feats, we certainly take off our hat to *our* practical man.—"Engineering," April 18, 1929.

WHAT CAUSES BALL BEARING TROUBLES?

Most Troubles with Ball Bearings Are Caused by Not Following the Directions for Mounting and Care that Are Given by the Bearing Manufacturer.

By Asher Golden, R B F Ball Bearing Co., New York City.

The directions for the mounting and application of ball bearings that are furnished by the bearing manufacturers give the user the necessary information as to what to do and what to avoid in order that bearings may function properly at maximum efficiency and live to a good old age. Many users, however, do not take the time to read these directions, because they believe that they know how to apply and use ball bearings. As a result, if

when mounting a bearing, one of the rings cracks, the user is likely to jump at the conclusion that the bearing is defective. If a bearing has been in use but a short time and becomes rough and noisy, no attempt is made to determine the cause, but it is assumed that the bearing is defective and it is returned to the manufacturer. As a matter of fact, if a bearing fails in any way whatever, it is assumed that the bearing is defective, and no thought is given to the possibility that the trouble may be due to the fact that the directions given by the manufacturer were not followed.

This attitude creates a serious problem for the bearing manufacturer whose product is unjustly condemned; it also causes expense and loss to the user of ball bearings who installs them in machines that he subsequently sells, and whose product, therefore, with improperly installed bearings, may be condemned by the ultimate consumer.

A SMALL SPECK OF DIRT OFTEN IS THE SOLE CAUSE OF BEARING TROUBLES.

A case came to the writer's attention recently that illustrates this point. A manufacturer of machine tools returned a considerable number of bearings as defective. He claimed that the bearings were of poor quality and that in consequence he had received many complaints from his customers and been put to considerable expense as well as suffering a loss of prestige. The bearings were carefully examined and the failure correctly diagnosed as being due to dirt.

At another time a user complained that in a lot of bearings received the holes were of different sizes. Some were correct, while others were too large or too small. It had never occurred to the person who found difficulty in assembling the bearings that the fault might lie with the spindles to which the bearings were fitted. This was suggested, and nothing further was heard of this complaint.

In another case, a user brought in a bearing that he claimed to be defective. He removed the bearing from a dusty box, held it by the inner ring with one hand and turned the outer ring slowly with the other. "Feel that," he said; "it has a high spot." We examined the bearing and found that the "high spot" was just a particle of dirt. The "high spot" was removed by simply washing the bearing with kerosene.

TO GIVE SATISFACTORY SERVICE, BALL BEARINGS MUST BE KEPT CLEAN.

I believe that bearing manufacturers would have fewer complaints and come nearer to the solution of this vexing problem if they attacked it backward—that is, started with the effect and went back to the cause. The directions might read, for example: "If on removing a bearing that has been in service but a short time it is found that the surface of the inner ring adjacent to the shaft is partly or wholly polished, then the bearing has worked loose on the shaft and the shaft has been turning in the inner ring; dirt may have caused this condition by working its way into the bearing and locking or jamming it. Do not expect a bearing to give satisfactory service unless it is kept clean."

I believe that the effects of abuses of ball bearings in one form or another might be all referred to in a pamphlet, which could be sent to a user who complains of defective bearings, and his attention could be directed to the particular paragraph that treats of the cause of the failure in his case. I believe that this would have a tendency to discourage unwarranted and unreasonable complaints.

When ball bearings came into use for bicycles, they were of the cup-and-cone type and rather coarsely made, in comparison with the precision bearings now manufactured for automobiles and machinery. No special direc-

tions were necessary for mounting and using bearings of the cup-and-cone type. Dirt in the bearing or some extra pull on the wrench in adjusting it did not seriously affect its operation. With the advent of the precision bearing, however, it became necessary and important to instruct the user how to fit and mount it and how to take care of it. The fit between the balls and the raceways of such bearings is so close that dirt seriously affects the operation.

BALL BEARINGS ARE SUBJECTED TO RIGID TESTS AND INSPECTION.

Owing to the extreme care that leading ball bearing manufacturers take in putting their product through every conceivable test to disclose defects before the bearings are shipped, it is safe to say that not more than one bearing in a thousand that is alleged to be defective has anything the matter mounting or abuse in service.

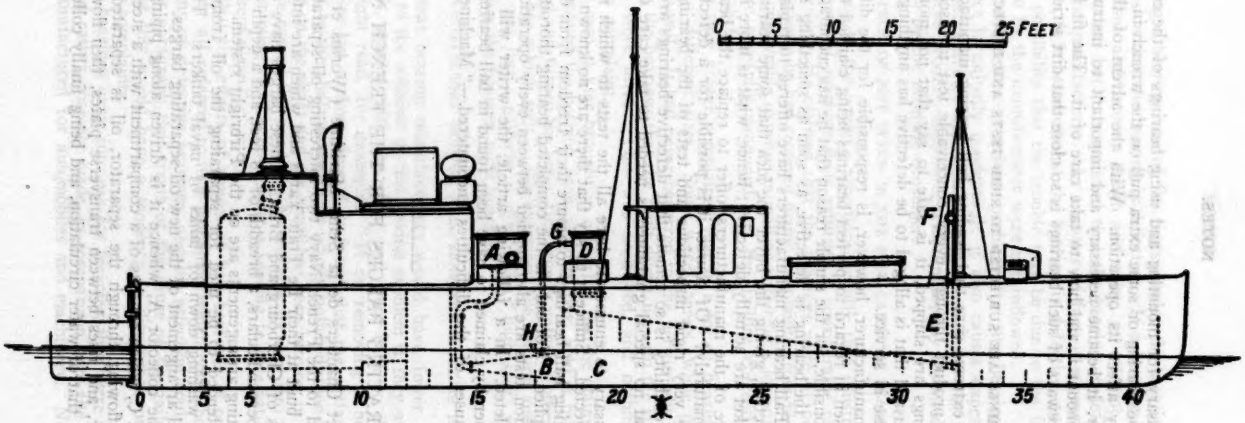
The bearing manufacturer, however, is responsible for the difficult problem that has arisen in regard to perfect bearings being claimed to be defective. He is responsible, for the simple reason that he has encouraged the user to complain that the bearing is defective, as soon as something seems to go wrong with it. Ball bearing manufacturers have offered to replace bearings that proved defective, giving the buyer the idea that some bearings that get out into the market are actually defective; hence, what is more natural than to take advantage of the manufacturers' offer to replace the bearing whenever it fails prematurely? Of course, it is possible for a defective bearing to pass even the very rigid inspections and tests of the bearing manufacturer, but the probability is so remote that defective bearings are extremely rare—so rare that no special guarantee is needed to take care of this condition.

It is not necessary to enumerate here all the tests to which modern ball bearings are subjected. Suffice it to say that there are no known methods of inspecting or testing ball bearings to assure their freedom from defects, that are not being applied. Not only is the completed bearing thoroughly tested, but these inspection tests are also applied between every operation until the bearing is completed. In a following article, the writer will outline the defects most generally claimed to have been found in ball bearings and will point out the causes of the difficulties encountered.—"Machinery," April, 1929.

OIL-SEPARATING BARGES FOR THE FRENCH NAVY.

The Ateliers et Chantiers de la Seine Maritime (Worms et Cie.) have recently delivered to the French Navy two interesting oil-separating barges, which have been built at their Le Trait Yards and which are intended to be used at the ports of Cherbourg and Brest. These barges have a length of 92 feet between perpendiculars, a breadth of 18 feet, and a depth of 9 feet.

The oil-separating arrangements are on the Pirbright system. The barges are primarily intended to be used for separaitng the oil from the water utilized for the washing down of tanks of naval tankers. The drawing shows the general arrangement of the new oil-separating barges. Oily water is admitted to the collector A, whence it is driven along piping B to the separator proper, C, which consists of a compartment with a steeply-inclined ceiling. While flowing through the separator, oil is separated from the water by gravity, and rises between transverse plates, then flowing in the opposite sense to that of water circulation, and being finally collected in the



ARRANGEMENT OF THE PIRBRIGHT TYPE OIL-SEPARATING BARGE FOR THE FRENCH NAVY.

tank D. On the other hand, water which has continued to run through the separator after being progressively separated from oil, is pumped overboard through the piping E. The tank D, into which oil is collected, is fitted with a float which is arranged so that it can float only in water and sinks in oil. When the amount of oil collected in the tank is sufficient the float sinks, then operating a lever which opens the oil valve G. Oil is then driven into the tank H through the pressure resulting from the difference of level between F and G.

Oil is progressively replaced by water in the tank D, until the float rises and closes the oil valve. It is arranged that water exhausted in the effluent pipe F contains less than 1/10,000th part of oil.

The recovered oil is further purified by means of centrifugal separators, with the result that the oil finally obtained contains less than 1 per cent of water. The extensive trials which have been carried out by the French Navy at Cherbourg and Brest have been very satisfactory. Successive applications were made to:—

1000 tons of water which had been used for washing the tanks of a tanker; a mixture containing 20 tons of fuel oil and 80 tons of water; and a mixture containing 40 tons of fuel oil and 10 tons of water.

From these trials it was found that water taken from the outlet of the separator contained only from 2 to 7/100,000th parts of oil, while fuel oil recovered after passing through the centrifugal separators contained only 0.5 per cent of water.—"Shipbuilding and Shipping Record," March 7, 1929.

THE NAVY ESTIMATES FOR 1929.

The Navy Estimates for the financial year 1929 (1st April, 1929, to 31st March, 1930), which were issued by the Government on the 8th March, once more contain no surprises for shipbuilders. The net total of the Estimates for the year is £55,865,000, which is a reduction of £1,435,000 on those for 1928. Apart from the fact that this reduction is somewhat greater than seems to have been expected, the Estimates present few specially noteworthy features, based strictly as they are on a naval policy adopted in 1925, when the Program of New Construction was announced to Parliament.

According to the explanatory memorandum by the First Lord of the Admiralty (the Rt. Hon. W. C. Bridgeman) which accompanies the Estimates, the amount proposed to be voted for new construction under that Program in 1929, is £8,621,626, or £1,008,231 less than in 1928. The main reason for this smaller provision is, of course, the cancellation of the three light cruisers from the 1927 and 1928 Program, as the result of which only seven cruisers under construction will be at the stage of requiring large payments in 1929, as against 10 in the previous year and 12 in 1927. Other contributory causes are the adoption of new designs, which have reduced the cost of the flotilla-leader and submarine depot ship included in the 1928 Program, and the fact that the progress during the year on new construction as a whole has been estimated on a very conservative basis—a policy which has been found to be conducive to economy in previous years.

"In preparing these Estimates," the First Lord adds, "we have, as in previous years, followed the plan adopted by the late Government in preparing the Navy Estimates for 1924; that is to say, after allowing for all known variations in expenditure, a considerable allowance has been made for possible under-spendings through unforeseen causes interfering with the progress of contract work. The allowance made on this account in 1929 is the same as has to a large extent justified itself in 1927 and 1928, but it is

necessary to repeat the reservation which I made in both these years as to the possibility of a Supplementary Estimate if no interference with the progress of work should in fact occur."

TECHNICAL PROGRESS.

Catapults of different types for the launching of aircraft, we are told, are now on trial and in process of development.

The progress of research has been satisfactory. While the chief object is improvement in offensive and defensive material, much of the work done is directed towards the reduction of expenditure by increasing the useful life of material.

The reduction of weight and the improvement of economy and reliability of engines continue to advance with each new design.

Attempts are being made to arrest the corrosion of steel condenser doors by coating them with rubber—a new process developed in this country.

A method of X-ray examination, developed by the Research Department of the Royal Arsenal at Woolwich, has been utilized with great advantage in the investigation of defects due to internal flaws in castings.

The development of heavy-oil engines for naval purposes has proceeded steadily at the Admiralty Engineering Laboratory. The limitations of existing designs have been further investigated and new proposals arising therefrom are under consideration.

The Government is keeping in touch with the progress being made in the use of pulverized fuel, and with the performances of working installations both ashore and afloat.

The various processes for the low-temperature distillation of coal for the production of fuel, lubricating and other oils are also being closely followed.

The development of means for preventing the pollution of harbors by the use of oil separators continues to be energetically pursued. Plants are now installed in barges at Portsmouth and Devonport, and the future action to be taken in this matter depends upon the results achieved with these installations.

Experiments are also in progress with a view to rendering serviceable the residual oils remaining in fuel tanks, etc. An Admiralty design of separator appears to be satisfactory, and should achieve important economies if successful.

SHIPS DELIVERED AND UNDER CONSTRUCTION.

The five 10,000-ton light cruisers of the *Kent* class (1924 Program), three of which were built in H.M. Dockyards, have successfully completed their trials and are now on service. The *Cornwall* (built at Devonport Dockyard and engined by Messrs. William Beardmore & Co., Ltd.), during the long voyage *via* South American ports to China, acquitted herself well, though very rough weather was at times encountered, and all these ships have proved themselves to be very good and economical steamers.

Of the four 10,000-ton light cruisers of the *London* class (1925 Program), the *London* is completed, and the *Devonshire* and *Sussex* are approaching completion. The *Shropshire* was launched on the 5th July last, and is expected to be completed in September next. The *London* was built at Portsmouth Dockyard, and engined by the Fairfield Shipbuilding & Engineering Co., Ltd.; the *Devonshire* at Devonport Dockyard, with engines by Messrs. Vickers-Armstrongs, Ltd.; the *Sussex* was built and engined by Messrs. R. & W. Hawthorn, Leslie & Co., Ltd.; and the *Shropshire* by Messrs. William Beardmore & Co., Ltd.

The two light cruisers of the *Dorsetshire* class (1926 Program) are progressing satisfactorily. The *Norfolk* was launched in December last by the Fairfield Company (who are also supplying the propelling machinery), and she is expected to be completed in May, 1930. The *Dorsetshire* was launched on the 29th January at Portsmouth Dockyard, and is being engined by Messrs. Cammell, Laird & Co., Ltd. She is expected to be completed in the summer of 1930.

Of the *B*-class cruisers (1926 and 1927 Programs), the *York* (built and engined by Palmers Shipbuilding & Iron Co., Ltd.), was launched on the 17th July last, and is expected to be delivered in March, 1930. The *Exeter*, which was laid down at Devonport Dockyard in August last and is being engined by the Parsons Marine Steam Turbine Co., Ltd., will be launched in the summer. The leading particulars of the *York* are given in the Estimates as follow:—

Length, feet and inches	540.0
Breadth, feet and inches	57.0
Mean draught at standard displacement, feet and inches	17.0
Standard displacement, tons	8,400
Estimated horsepower on full-power trial	80,000
Corresponding speed at standard displacement on full-power trial, knots	32¼
Total oil-fuel capacity, tons	1,900

The armament, excluding machine guns and torpedoes, consists of six 8-inch, four 4-inch anti-aircraft and four 3-pounder guns, and two 2-pounder pom-poms.

The two light cruisers of the 1928 Program, which will be of the 10,000-ton type, are being ordered from Portsmouth and Devonport Dockyards respectively. They had not been laid down at the time the First Lord's statement was prepared, nor had any announcement been made as to the placing of the contracts for their propelling machinery, as provision was taken in the Estimates for ordering the ships in March of this year.

As regards torpedo-boat destroyers, the extended cruise of the *Amazon* (built and engined by Messrs. John I. Thornycroft & Co., Ltd.) and *Ambuscade* (Messrs. Yarrow & Co., Ltd.) in South American waters afforded the opportunity of testing their equipment and the large number of new features embodied in their machinery installations. During this period of three months they steamed a distance of 10,000 miles without developing any defects other than such as are normally undertaken by the ships' staffs. The eight destroyers of the *Acasta* class (1927 Program) are proceeding satisfactorily, and it is anticipated that seven of them will be launched and completed during the financial year 1929. The remaining vessel of the class—the *Acheron*, which is being built by Messrs. Thornycroft to the order of the Parsons Company—is being fitted with high-pressure steam machinery, and is not expected to be completed until the summer of 1930. The destroyers of the *Acasta* class (including the *Acheron*) are each 312 feet by 32 feet 3 inches by 8 feet 6 inches mean draught at a standard displacement of 1330 tons. The armament includes four 4.7-inch guns; and the propelling machinery, to develop 34,000 H.P. on full-power trial, is expected to give each vessel a speed of 35 knots. The flotilla-leader *Codrington*, ordered from Messrs. Swan, Hunter and Wigham Richardson, Ltd., at the same time as the destroyers of the *Acasta* class were placed, is 332 feet by 33 feet 9 inches by 8 feet 10 inches mean draught at a standard displacement of 1,520 tons. Her propelling machinery, by the Wallsend Slipway and Engineering Co., Ltd., will develop 39,000 H.P., giving 35

knots, and she will mount five 4.7-inch guns. Orders for the flotilla-leader and eight destroyers of the 1928 Program have been placed with the contractors whose names have already been published in this journal.

As to submarines, the *Osiris* (Messrs. Vickers-Armstrongs, Ltd.) is now completed, and the remaining vessels of the *Odin* class (1926 Program) will be completed at intervals through the summer of 1929. The six submarines of the *P* class (1927 Program) are progressing well, and it is anticipated that all will be launched during this calendar year and that they will be completed during the financial year 1929. Of the submarines of the 1928 Program, orders for five to be built by contract have been placed with the three firms whose names were given in the March Number of this journal, while one vessel will be built at Chatham Dockyard.

The submarine depot ship *Medway* (built and engined by Messrs. Vickers-Armstrongs, Ltd.) will proceed on her trials about April, 1929, and be completed shortly afterwards. This vessel, as our readers are aware, has heavy-oil engines of a commercial type. The submarine depot ship of the 1928 Program was originally intended to be built by contract, but it has since been decided to build her at Chatham Dockyard.

The fleet repair ship *Resource* (built and engined by Messrs. Vickers-Armstrongs, Ltd.) has been launched, and will probably be completed in the autumn of 1929.

The first two sloops of a new design—H.M. ships *Bridgewater* and *Sandwich*, built and engined by Messrs. R. & W. Hawthorn, Leslie & Co., Ltd.—are nearing completion. The four sloops of the 1928 Program will shortly be begun. The construction of these was originally allocated to contract, but it has since been decided to build two of them at Devonport Dockyard, while the remaining two have been placed on the Tyne, as announced in our columns last month. The river gunboat of the same Program will be built by contract.

REFITS AT THE ROYAL DOCKYARDS.

The *Revenge* completed a large refit in January, and has rejoined the Mediterranean Fleet. The *Iron Duke* will complete a large refit in May. The *Hood* and the *Benbow* will commence large refits during 1929. The bulging and reconstruction of the *Malaya* are now complete, and she will rejoin the Mediterranean Fleet shortly. The *Valiant* was taken in hand for bulging last month (March). The reconstruction of the *Courageous* as an aircraft-carrier was completed in May, 1928; while the reconstruction of her sister ship—the *Glorious*—is expected to be completed in the latter part of 1929. The final repairs following the stranding of H.M.S. *Dauntless*, off Halifax, are being executed at Portsmouth Dockyard, preparatory to her rejoining the Fleet.

NEW CONSTRUCTION UNDER THE 1929 PROGRAM.

The new Program for 1929 includes provision for commencing the construction of the following ships:—Three light cruisers, one flotilla-leader, eight destroyers, six submarines, one net-layer and target-towing vessel, and six sloops. Two cruisers, two submarines, the net-layer, and two sloops will be built in H.M. Dockyards, while the remaining vessels will be constructed by contract.

The sloops are additional to the Program as set forth in Command Paper 2,476, dated 27th July, 1925. On the other hand, the aircraft-carrier included in it as to be commenced in 1929 will not be proceeded with this

year, as it is desired to obtain more experience with the *Courageous* and *Glorious* before laying down another large aircraft-carrier.

The provision made for expenditure during the financial year 1929 on the vessels to be begun under the new Program is very small. For the cruiser to be laid down at Portsmouth Dockyard, it amounts to £7,564; for the cruiser at Devonport Dockyard, £7,678; for the two submarines at Chatham Dockyard, £20,028; for the two sloops at Devonport Dockyard, £2,410; and for the net-layer and target-towing vessels at Chatham Dockyard, £1,564. The contract-built vessels are similarly treated. For the cruiser to be laid down in a private yard, £10,480 is provided; for the flotilla-leader and eight destroyers, £2,954; for the four submarines, £3,490; and for the four sloops, nothing.

EXPENDITURE ON DOCKYARD-BUILT SHIPS.

For expenditure on the light cruiser *Dorsetshire* during the financial year 1929, £670,754 is provided, including £38,723 for contract work in connection with the propelling and other machinery, and £234,143 for gun mountings, torpedo tubes, etc. For the cruiser to be laid down at Portsmouth Dockyard under the 1928 Program the corresponding sums are £421,097, £56,100 and £81,195; while for the cruiser to be begun at Devonport Dockyard the amounts are £420,905, £56,100 and £81,915.

The sum provided for expenditure during the financial year 1929 on the cruiser *Exeter* is £572,223, including £119,835 on propelling and other machinery, and £167,151 on armament.

For expenditure on the submarine *Odin* £89,763 is provided; on the *Parthian*, £18,906, and on the submarine to be laid down under the 1928 program, £147,630. All these three submarines are at Chatham Dockyard.

For the submarine depot ship to be begun at Chatham under the 1928 Program £172,803 is allocated; while on the two sloops to be built at Devonport Dockyard under the same Program £144,672 will be expended.

EXPENDITURE ON CONTRACT-BUILT SHIPS.

On the cruiser *Shropshire* £363,354 is to be spent in the financial year now commencing, of which £170,230 is for contract work in connection with the hull, fittings and equipment, £53,607 is for propelling and other machinery, and £86,110 for gun mountings, torpedo tubes, etc. For the cruiser *Norfolk* the corresponding figures are £657,517, £273,070, £62,394 and £217,418; and for the cruiser *York*, £434,455, £143,000, £68,162 and £137,893.

For the flotilla-leader *Codrington*, the total expenditure during the year is estimated at £149,703; for the two destroyers *Acasta* and *Achates* (Messrs. John Brown & Co., Ltd.), £328,013; for the destroyer *Acheron*, £166,228; for the two destroyers *Active* and *Antelope* (Messrs. R. & W. Hawthorn, Leslie & Co., Ltd.), £264,291; for the two destroyers *Anthony* and *Ardent* (Scotts' Shipbuilding & Engineering Co., Ltd.), £264,352; and for the destroyer *Arrow* (Messrs. Vickers-Armstrongs, Ltd.), £147,111.

The total amount allocated for expenditure during the new financial year on the flotilla-leader and the eight destroyers to be begun under the 1928 Program is £553,922.

As regards the submarines under construction in private yards, £226,930 is to be spent during the year on the *Olympus* and *Orpheus* (Messrs. William Beardmore & Co., Ltd.); £124,377 on the *Osiris*, *Oswald* and *Otus* (Messrs. Vickers-Armstrongs, Ltd.); £677,165 on the *Perseus*, *Proteus*, *Pandora* and *Poseidon* (Messrs. Vickers-Armstrongs, Ltd.); £183,732 on

the *Phœnix* (Messrs. Cammell, Laird & Co., Ltd.); and £340,693 on the five submarines ordered under the 1928 Program—three from Messrs. Vickers-Armstrongs, Ltd., and one each from Messrs. William Beardmore & Co., Ltd., and Messrs. Cammell, Laird & Co., Ltd.

A sum of £103,869 will be expended during the financial year 1929 on the submarine depot ship *Medway*, and £300,174 on the repair ship *Resource*.

On the two sloops ordered from Messrs. Swan, Hunter and Wigham Richardson, Ltd., under the 1928 Program £72,938 will be expended in 1929, while on the gunboat to be begun under the same Program the sum to be spent during the financial year is estimated at £39,035.—"The Shipbuilder," April, 1929.

ELECTRON.

By Ing. A. W. Bonaretti.

Just as the nineteenth century properly considered is the century of coal, iron and steam, so the twentieth century appears thus far to be characterized by marvellous developments in electricity and chemistry, the essential elements of modern engineering and hence of the very structure of modern industrial economy.

This new orientation of technical engineering is particularly important to our country because electro-chemistry and electro-metallurgy, by making it possible for Italy economically to produce light metals capable of replacing iron and other heavy metals in many cases with advantage, will enable us appreciably to diminish the industrial inferiority created here by nature, which made us poor in iron and still more poor in coal.

Aluminum and magnesium are light metals which chemistry and electricity enable us to extract from our rocks by modern processes now highly perfected and economic; they are the essential elements of two series of alloys having physical and mechanical characteristics of the highest importance in engineering and especially in modern mechanics. Not so long ago aluminum was considered as a precious metal; in 1855, at the exposition of Paris, some small blocks of aluminum were exhibited not far from the crown jewels, and quite properly so because the cost of these aluminum blocks was estimated at about 3000 lire in gold per kilogram.

Still more rare and costly at even a more recent date was magnesium, which moreover found useful employment only in certain pharmaceutical products and in the preparation of fireworks.

But it was not alone the high cost of production which prevented and limited the use of these metals; they also lacked in large measure the necessary mechanical characteristics and some of them also the desired physical and chemical properties. This deficiency has been corrected recently, on the one hand by the adoption of electro-chemical methods of production which enormously reduces the initial cost of these metals, and on the other hand by alloying them with others and thus obtaining combinations of great industrial importance capable of employment to a degree which is already very extensive and which will be further increased by improvements in the alloys themselves, their heat treatments and the methods used in working them.

It should be borne in mind that the knowledge of these metals and their alloys is too recent not to leave room confidently to expect further technical and economic improvements and that we shall soon see day by day a greater demand for them in modern construction. Everything that is intended to

be transported or moved demands lightness, because lightness means convenience and economy, the principal conditions characterizing modern activity never sated with speed, never rewarded with too high outputs.

The proper construction of the whole and the correct-designing of details, as well as the adoption of metals and especially of steels of constantly increasing strength made it possible thus far to satisfy this demand for lightness. Today, it seems that further progress is attainable in this direction by the use of the modern light alloys (of aluminum) and very light alloys (of magnesium) wherever such use can be made rationally, that is with a proper consideration of all the special characteristics of these alloys and with the proper adaptation of all the practical conditions relative to the use it is intended to make of them.

Of these alloys, the series derived from aluminum is well known and much used in Italy; less well known is the series derived from magnesium which appeared on the Italian market about eight years ago as an importation from Germany and today is beginning to be used rather extensively, especially in the works of the Fabbrica Automobili Isotta Fraschini, which more than five years ago established a foundry for light metals of which it makes extensive use in its aviation engines, automobiles, and mechanical constructions of all kinds.

Of the brilliant results obtained in this field of its activity, the Isotta Fraschini provided a clear and effective illustration at the International Metallurgical Exposition held at Paris in 1927. At its stand were exhibited divers parts of machinery and pieces of the most varied dimensions, all of electron, interesting to aeronautics and optics, the textile industry and naval constructions, the automobile industry and electro-mechanic and telephonic constructions, which called forth the warmest admiration and praise from the most highly reputed European metallurgists who proclaimed electron to be "a sovereign alloy" owing to the great diversity of applications that could be foreseen for it, entitling it to the highest award from the Committee of the Exposition.

Then at the Milan Fair, 1928, the electron stand of the Isotta Fraschini attracted the universal attention of technicians and was visited by a large number of foreigners and nationals.

Magnesium is a ductile and malleable metal, silvery in appearance, perfectly stable in dry air and resistant to the action of alkalies; it dissolves in dilute acids developing hydrogen, after much experimentation and many ups and downs, its metallurgical technique was finally established in 1896 in the Hemelingen Establishments near Bremen and shortly afterwards in the works of the Griesheim Electron of Bitterfeld, where are now produced not less than 200 tons of magnesium a month.

Magnesium possesses the following principal characteristics:

Atomic weight	24.3
Density at 15 degrees	1.72
Melting point, degrees	651
Boiling point, degrees	1120
Coefficient of elongation 10^6	25
Specific heat between 15 degrees-100 degrees	0.248
Heat conductivity	0.38
Specific resistance at 20 degrees $\times 10^6$	2.9
Tensile strength of ingot metal kilogram	9

In Germany magnesium is obtained by electrolysis from the carnallite of Stassfurt (bichloride of magnesium and potassium) and, freed from the impurities which accompany it and in particular of the chlorides which greatly

decrease its resistance to atmospheric agents, is placed on the market with reservations as to its content of impurities of from 0.4 to 0.8 per cent of potassium and of 0.1 per cent of silicon, iron, aluminum and calcium as a whole.

In Italy magnesium may be obtained by electrolytic processes from dolomite (bicarbonate of calcium and magnesium) which is found in enormous deposits in the mountainous portions of our country.

Magnesium by itself would not be used in mechanical constructions owing to its lack of resistance and the readiness with which it is corroded by water; it finds an enormous use when combined as an alloy with other metals because it then acquires noteworthy mechanical characteristics and an entire, or at least adequate resistance to atmospheric agents and to the action of alkalies, greases, etc.

A large number of alloys of magnesium are made with a view to obtaining products which have a wide and important use in modern mechanical constructions. Conspicuous among all and by far the best known at the present time in Italy is the German series which goes under the name of *electron* and comprises alloys that can be fused, stamped, rolled, drawn, etc.

Electron alloy intended for the preparation of castings is distinguished from the others of the German series by the mark AZG and is composed of:

Magnesium, per cent	90.5
Aluminum, per cent	6
Zinc, per cent	3
Manganese, per cent	0.5

Different proportions of the components which are sometimes reduced to three or only two, give different characteristics to the alloys of this series and their suitability for the uses mentioned above.

Electron is now supplied in castings from the smallest to the largest dimensions in stamped parts, in shapes of the most varied section and form and with tolerances of half a tenth of a millimeter in rolled parts of all commercial sizes, in sheets of soft, semi-hard and hard quality, in tubes of all diameters up to 100 millimeters and of thicknesses down to a single millimeter and also in fine wire. The principal mechanical and technical characteristics of electron are the following:

Specific weight	1.75-1.85
Modulus of elasticity	4000 to 5000 kg.-mm^2
Elongation	5-18 per cent
Reduction in area	9-35 per cent
Tensile strength	17-40 kg.-mm^2
Elastic limit	5-25 kg.-mm^2
Yield point	9-30 " "
Breaking load under compression	30-45 " "
Brinell hardness	43-90
Resilience	30-140 kg.-cm.-cm^2
Rupture	13-18 kg.-mm^2
Limit of endurance to rotary flexures of twisting moment	5.5-13
Melting point	635 degrees-650 degrees
Coefficient of dilatation under heat	0.000023-0.000027
Shrinkage (in ingots)	about 1.3 per cent
Specific heat	0.24
Heat conductivity (about 1/3 of that of copper)	0.32-0.38
Electric conductivity (about 1/3 of that of copper)	16-22
Antimagnetic in all qualities	

In this table the lower characteristics were obtained from cast pieces and the higher ones from shaped, rolled and stamped pieces after being subjected to a proper heat treatment.

Electron has a silvery white appearance similar to that of aluminum and after working presents beautiful reflecting surfaces. Exposed to the air it slowly becomes covered with a light oxide film which in very many cases constitutes a protective covering against further oxidation.

Electron, in contrast to aluminum, is insensitive to the action of alkalies and their aqueous solutions, and is unaffected by grease, oil, petroleum and benzine.

The quaternary alloy AZG is practically insensible to the action of fresh water and is sufficiently resistant to that of salt water in the presence of which it behaves in a similar manner to silumin and aluminum as will be seen further on.

Electron may be cast in earth and in chills under pressure. The casting of this alloy, however, requires proper methods and care, and special and painstaking skill which can be satisfactorily acquired only by long practice.

The great chemical affinity of magnesium for the oxygen in the air is the essential cause of the difficulty in casting this metal and the reason why special watchfulness must be exercised to prevent the burning of the metal during smelting and pouring as well as very meticulous and painstaking care to guarantee absolute metallic continuity and perfect compactness of the castings. When the proper care has been exercised in making series castings, the same characteristics can be obtained in every way that are obtained in the small test bars cast expressly for the purpose, because in the mass of the cast pieces it is possible to avoid with certainty failures of continuity through microscopic cracks, porosity, and the presence of small cavities which it is practically impossible to avoid in the casting of aluminum.

This peculiar characteristic of electrons has the result that the practical resistance of electron castings is superior to that of aluminum castings by an amount much greater than the simple ratio of the respective coefficients of mechanical resistance.

In order to represent in a clear and simple manner the value of these characteristics of electron in comparison with aluminum, it will be sufficient to point out that two castings, one of electron and the other of aluminum, identical in form and thickness, would be characterized respectively by the following figures:

	Aluminum	*Electron*
Weight	100	68
Practical resistance	100	135

The excellent properties possessed by electron, as shown even by the above hasty survey, would lead us to believe that its introduction would be extremely easy and its applications very extensive especially when it is realized that with the same tensile strength of cast, stamped and rolled pieces, electron is, as a rule, no less economical than aluminum, and that even if there were an economic difference in favor of aluminum, this difference is slight and is more than amply compensated for by the indirect benefits resulting from the marked saving in weight realized by the use of electron.

In actual practice, this case of introduction and extensive application has not been realized; only in recent times through the magnificent example of the wide and various application made of this metal by the Isotto Fraschini

has electron given signs of being received without too much hesitation and without serious doubts by the different industries.

Three obstacles, besides the natural resistance to the adoption of anything new and not yet sufficiently tested, have contributed and still contribute to this difficulty of introduction:

The doubt as to whether electron may not be inflammable like magnesium;

The doubt as to whether its working it may not prove to be difficult, expensive and dangerous;

The doubt as to whether it possesses sufficient stability under the corrosive action of fresh water, salt water and steam.

Doubt as to the inflammability of electron may be due to the record of free magnesium for conspicuous chemical activity, has no foundation as regards electron in the forms and dimensions adopted in actual practice. For more than five years the Isotto Fraschini has been working electron on machine tools of the most varied type, of which it possesses a great number in its shops, for the rough and worked parts used extensively in the manufacture of its automobiles and marine and aviation engines, has never felt the need of precautionary measures as protection from possible fires due to or facilitated by electron.

The possibility of the combustion of electron during its working may arise only when the tool is either badly designed or badly fitted, because then the material may be carried off in powdered form and highly heated in the process of working. This case, which would not in fact arise if the workman was performing his work with proper technical skill does not represent anything of a serious or dangerous nature in any event, because the combustion is limited merely to the operation in hand and a little intelligence will enable the incipient conflagration to be immediately quenched.

With the exception of this case, which occurred on a few rare occasions in the beginning in the shops of the Isotto Fraschini, electron is inflammable only when it remains in a state of fusion, a condition which, it is true, constitutes one of the principal difficulties in the work of smelting, but which never arises in its practical employment. Moreover, recent experiments carried out for military purposes confirmed the perfect safety of electron as regards its inflammability in connection with the firing of incendiary projectiles.

As regards the possibility of working electron, it may be said that it lends itself splendidly to any sort of working with a high degree of ease, certainty and economy in time and in the power required in the machine tool, while its extreme lightness makes its transportation easy and rapid. Working with the saw, lathe, plane, milling and boring machines, dressing, punching and polishing machines is all executed dry and is always very economical.

Electron does not, like aluminum, have a tendency to *catch* in the tool; the worked parts have a smooth, polished, reflective surface; threaded parts present and keep their surfaces with well defined edges which are resistant and perfect. The exceptional advantages of easy, reliable and economic working possessed by electron are superior to those of any aluminum alloys by at least 100 per cent, provided that, as was the case in the Isotto Fraschini workshops from the beginning machinery is adapted to velocities higher than those commonly used and the machines fitted with tools having a profile with the proper angle of incidence, attack, discharge, etc.

Various designers of machine tools, among them one English and one German designer, both world famous, have already designed and placed in

the market, automatic machines making 5000 revolutions per foot for working electron pieces in large series, machines whose very existence demonstrates the importance of these alloys and the economy that may be attaind in their working.

To give a rough idea of the great difference between the speeds necessary for working iron, pig iron, steel, etc., and those for electron, it suffices to say that the latter may be about of the order used for working wood. The saving in power and time attained in working electron is so notable as to make it economically desirable to replace objects, pieces and parts of machinery made of iron, pig iron or steel, etc., with other identical ones made of electron, when, speaking roughly, the cost of working the piece in heavy metal is not less than that of the rough piece itself.

Another doubt which in the past has justly caused apprehension in the technical world, but which should be abandoned for the future, especially in view of recent advancements made in the chemical composition of electron alloys, is that of the instability of this metal in the presence of fresh water, steam and sea water. This grave defect did in fact exist in electron until about two years ago, to such an extent as greatly to limit its field of application, but has now been so reduced that the use of electron in nautical constructions may and should be largely developed.

In the thorough and rigid tests conducted through months at a time by the Isotto Fraschini by placing cast, rolled and stamped pieces of their A.Z.F. electron alloy of the most varied sizes in fresh water, exposed to the weather, in salt water completely submerged, awash or completely emerged but surrounded by sea air, and in the numerous experiments made by an English firm of world-wide fame as the makers of aviation engines, with the standard English aluminum alloy L_5, the good behavior of A.Z.F. electron was demonstrated, and it may be regarded as quite as insensible to attack by fresh or salt water as the English L_5 aluminum alloy. Furthermore, the liability of electron to attack by fresh and salt water may easily be overcome by coating it with special varnishes which, as has been confirmed by long experience by the Isotto Fraschini, adhere perfectly to this metal and thus accomplish their protective work in a very durable and complete manner.

It is furthermore opportune to call attention to the fact that studies are now in progress intended to offer engineering of the present-day structural alloys of greater resistance to the action of the atmosphere, fresh and salt water and steam, and that these studies are leading to results that are constantly more noteworthy. The difficulties inherent in the use of electron for framework have been studied and overcome, it being possible to unite profiles of the most varied dimensions in a perfectly satisfactory manner so as to meet all technical requirements by means of rivets made of a special alloy of aluminum and magnesium, $M.G_5$. The same is true of autogenous welding of electron plates of $A.M._{05}$ alloy, and electric welding with special apparatus.

The applications of electron are already very numerous. *In aeronautical constructions* the casings of the engines and many cast parts of the same are almost always made of electron; frequently the wheels of flying machines and other details of their structure, sometimes the tanks, sheathings, instrument boxes, and certain parts of the military armament, etc., are made of this metal.

In automobile manufacture electron is employed in the cast parts of engines, in the casings of the gear shifts, etc., especially of auto busses and trucks.

Applications of considerable importance are now being made of electron in railway and tramway constructions in which lightness means economy of motive power, more rapid starting and stopping, less rapid wear of tracks and roadways and of the vehicles themselves.

This very light alloy is used in no less important constructions of an electro-mechanical nature (protective casings of electric motors, fans for automatic ventilation, supports, etc.), in optical and telephonic instruments, in the construction of machines for spinning and weaving, in naval construction, and in very many other devices in which lightness has a logical importance.—"L'Ala d'Italia," March, 1929.

BOOK REVIEWS.

STEAM TURBINES. A PRACTICAL AND THEORETICAL TREATISE FOR ENGINEERS AND STUDENTS, BY JAMES AMBROSE MOYER, S.B., A.M. Sixth Edition. Revised and Enlarged. Published by John Wiley & Sons, Inc., New York. Price, $4.50 net. 557 pages.

In addition to the subject-matter contained in the Fifth Edition, the present volume discusses at greater length the application of the bleeder turbine and the design of turbine details, including the calculations for a reaction turbine.

Engineers have been familiar with the technical writings of Professor Moyer for over twenty years and will welcome the Sixth Edition of STEAM TURBINES, which is valuable to students as well as to design and operating engineers.

CAPTAIN O. L. COX, U. S. NAVY.

STEAM, AIR AND GAS POWER, BY WILLIAM H. SEVERNS AND HOWARD E. DEGLER. Published by John Wiley and Sons, Inc. List price, $4.00.

In this book are described typical examples of power apparatus such as boilers, turbines, reciprocating engines, air compressors, etc. Supplementing the descriptive matter, which forms the bulk of the book, is a brief mathematical outline of the basic underlying thermo-dynamic principles. Mathematical notations and methods follow the form used by Professor Goodenough. Of some 400 pages, approximately 136 are devoted to engines and their auxiliaries; another hundred to boilers and boiler room auxiliaries and accessories; while air compressors, internal combustion engines, and principles of thermo-dynamics occupy practically one-third of the book.

Due to the scope of the subject-matter, considerable care was necessarily exercised in selecting the subjects for discussion. It is noteworthy that the authors have eliminated wherever possible everything which could be left out without impairing the continuity. The illustrations are clear and well chosen. The book is easy to read and will serve as excellent preparation for the student who wishes a preview of engineering before commencing on regular academic courses. Taken in conjunction with various engineering handbooks, it should prove a valuable addition to an engineer's library.

LIEUT. COMDR. BOYD R. ALEXANDER, U. S. NAVY.

"THE STORY OF A NORTH SEA AIR STATION," BY C. F. SNOWDEN GAMBLE, XVI 446 pp., illustrated with plates and line drawings. Sketch Chart of North Sea. Published by Oxford University Press, American Branch, New York. Price, $7.50.

Although primarily the story of the British Naval Air Station at Greater Yarmouth, this interesting and distinctly readable book has a much wider scope than indicated by its title. A brief account is given, in the introduction, of the Royal Naval Air Service, from its birth as the "Naval Wing" of the short lived "Royal Flying Corps," until the Greater Yarmouth Air Station was commissioned in April, 1913. The development of this station, and of the British flying service as a whole is then carried up to the outbreak of the war.

Succeeding chapters take up the Naval Air Service in general; our particular station, and the opposing German stations, at the beginning of the war, and for each year thereafter, until the formation of the Royal Air Force in April, 1918, and the end of the war.

Familiar names are frequently met with, such as that of E. R. A., F. W. Scarff, one of the twenty-two individuals who were the nucleus of the service which in a few years was numbered in thousands. The "Scarff Ring Mount" is still the standard mount for free machine guns. It is interesting to note that the first dirigible experimented with by the British, the *Mayfly*, embodied a successful water-recovery apparatus.

An amusing tale relates how it came about that the Royal Naval Air Service adopted as its device a German eagle.

Throughout the book, the writer discusses the Germans without malice, and with the ungrudging admiration so commonly found among the flying personnel. At the hour of final surrender, they felt "* * * a sort of sneaking pity for the enemy who had fought the war in the air in the North Sea so cleanly and so gallantly."

The story of the defense against the air raids is most interesting. The feeling, almost of despair, which resulted when the first Zeppelin raiders returned to their bases, seemingly immune to British attack, whether by anti-aircraft fire or by airplane, gave way almost to hero worship, when once there had been mastered the technique of bringing down these vulnerable, hydrogen-filled aircraft. From the enemy point of view also, this story is well told.

The book is in no sense technical in its handling of material, but nevertheless contains much of interest to those concerned with aircraft and their armament. During the period of the war, airplanes developed from stick-and-wire affairs to the Curtis boats, and from their original armament of a sawed-off shotgun loaded with chain shot, grew the synchronized fixed machine guns, free Lewis guns and bombs, in much the form in which they are familiar to us today.

So much ground is covered that this account can not be considered complete. The chapter on enemy submarines adds nothing to the common knowledge of these craft, and is inaccurate in detail. It might have been omitted. The stories of individual actions and patrol experiences, add much to the value of the book and to the pleasure of reading it. It is a relief to read an air book in which airplanes are never referred to as "ships." We infer that the author has been to sea.

"The Story of a North Sea Air Station" is sufficiently shop to satisfy those who read for the good of their souls, and sufficiently readable to please those who desire a good yarn of carrying on over the North Sea.

LIEUT. COMDR. SHERWOOD PICKING, U. S. N.

ASSOCIATION NOTES.

The Society takes great pleasure in announcing the award by the President of the United States of the Distinguished Service Medal to Rear Admiral Chas. W. Dyson, U. S. Navy, retired. Admiral Dyson has been an active member of the Society since its foundation, is one of its past presidents and has been twice its prize essayist. The citation is quoted:

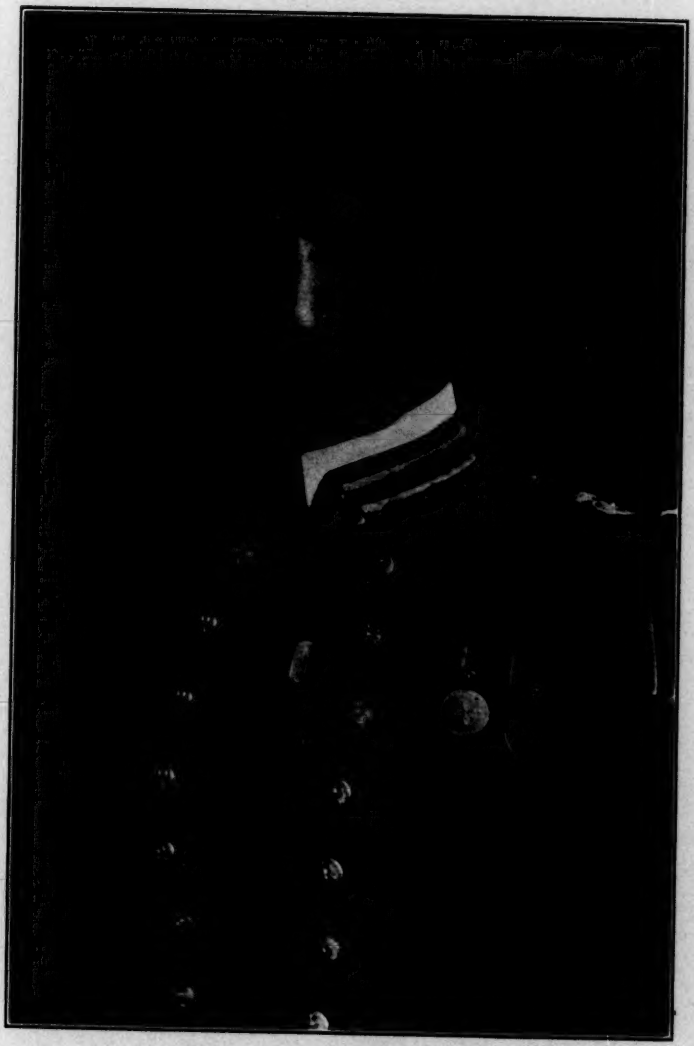
The President of the United States takes pleasure in presenting the DISTINGUISHED SERVICE MEDAL to

REAR ADMIRAL CHARLES W. DYSON, U. S. N., RETIRED

for service as set forth in the following

CITATION

"For exceptionally meritorious service to the Government in a duty of great responsibility in charge of the Design Division of the Bureau of Engineering and particularly for the design of the propellers of the aircraft carriers *Saratoga* and *Lexington* and of many other naval vessels which have been made by him personally and which have proven very successful on trials. Rear Admiral Dyson, as a result of his ability, study and application, is recognized as one of the foremost authorities in the world in the field of propeller design. During his long years of service in the Navy he has given it, without stint, the benefit of his knowledge and experience and the success of the machinery installations of many of our vessels is due to his thought and effort."



CHARLES W. DYSON
REAR ADMIRAL, U. S. NAVY, RETIRED

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Loeser, Arthur E., Ensign, U. S. N.

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